THE INFLUENCE OF GEOMETRY ON DRAGLINE BUCKET FILLING PERFORMANCE

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DECLARATION

I, the undersigned, hereby declared that the work contained in this thesis is my own original work and has not previously in its entirety or in part been submitted at any university for a degree.

Signature:. Date:

Summary

This thesis describes the procedure followed in order to establish geometry changes to the current lightweight dragline bucket built by Wright Equipment Company to improve its performance. The bucket performs very well as is and big changes were not expected. This project was done as part of the Mechanical Engineering Masters Degree requirements at the University of Stellenbosch, while being employed by Wright Equipment.

A scale model dragline was designed and built to be used in collecting the data. The most significant variables when considering dragline bucket filling were established and ranked according to their relative influence. The tests were done, using four different scale model dragline buckets and changing a number of variables on each of them at different drag angles and in different digging conditions.

Eventually it was found that a shorter, wider bucket with a lower hitch, resulted in improved performance as far as filling distance and filling energy requirements were concerned. The maximum required drag force was not increased, meaning stalling of the bucket would not be a problem. The changes do have some structural implications though and should be investigated before any changes are made.

Opsomming

Die tesis beskryf die prosedure wat gevolg is om geometrie veranderinge aan die liggewig sleepgraafbak, wat deur die maatskappy Wright Equipment vervaardig word, te ondersoek. Die uiteindelike doel was om die produktiwiteit van die bak (gedefinieer as die hoeveel bogrond wat in 'n gegewe tyd geskuif word) te verbeter. Die projek het deel gevorm van die vereistes vir 'n Meestersgraad in Meganiese Ingenieurswese by die Universiteit van Stellenbosch.

'n Skaalmodel sleepgraaf is ontwerp en gebou vir gebruik in die versameling van die nodige data. Die belangrikste faktore betrokke by die vulling van sleepgraafbakke is vasgestel en in rangorde van belangrikheid gegroepeer. Vier skaalmodel sleepgraafbakke, waarop verskillende geometriese veranderinge gedoen kon word, is gebruik in toetse teen verskillende sleephoeke en grondkondisies.

Daar is gevind dat 'n korter, wyer bak met 'n laer sleeppunt 'n verbetering in werkverrigting gee in terme van afstand om te vul en energie vereis. Die maksimum sleepkrag benodig is ook nie verhoog nie, met die gevolg dat stolling nie 'n probleem sal wees nie. Die voorgestelde veranderinge het egter strukturele implikasies wat eers ondersoek sal moet word voor enige veranderinge gedoen word.

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1. INTRODUCTION

Draglines are used extensively on open strip mines to uncover coal, which is one of the primary energy sources known to mankind. Removing the overburden from the coal is the most expensive and time consuming operation on coal mines. It is therefore very important that this stripping should be done as efficiently as possible.

South Africa and Australia are two of the worlds largest producers of coal. In South Africa it is generally accepted that an improvement of 1% in the efficiency of a dragline, will result in a R1 million increase in annual production per dragline. Additionally, the energy consumption of a dragline is vast and if any savings could be made on the amount of electrical energy consumed, it will result in bigger profit margins.

A lot of research has been done on improving the productivity of draglines. This involved work on systems to monitor production (section 3.4.2.2), mathematical models to predict productivity (section 3.4.2.1) and research on dragline operation (section 3.1). Very little work has been done on dragline buckets, but it is widely accepted that the bucket does have a big influence on the productivity of the dragline (Rowlands, 1991; Smit, 1996; Pundari, 1981 and Lumley and Jensen, 1996). Where research has been done by bucket manufacturers it was not published because of competition between the bucket manufacturers.

Making use of the knowledge of a bucket shop supervisor at Optimum mine, Dries Smith, a new bucket was developed by Northwest Applied Technology (design), VR-Steel (steel supply) and Barlows Equipment Manufacturing Company SA (manufacturers) who formed a joint company called Wright Equipment. This bucket proved to be very successful in the mining industry and when exporting the buckets to Australia became a real possibility, it was decided to initiate a bucket development program. For this purpose a scale model dragline was built and tests were conducted. The aim of the project was to establish a geometry for a dragline bucket that would allow it to fill in a shorter distance (in different materials) and with lower energy consumption than the current design. The maximum required drag force was not to increase as that could cause the bucket to stall. The project was therefore focused on increasing the productivity of a dragline by optimising bucket design. Huge profits in terms of increased production is possible as discussed in section 3.5.

In section 2 a short overview of strip mining is given and in section 3 a detailed discussion on draglines and operating methods associated with them is presented. These two sections were included as it was seen as important to have a thorough knowledge of especially the operation of draglines and at least an idea of the operation and layout of a coal mine. They also show draglines to be the most popular stripping machine and shows that huge financial gains in terms of increased production is possible if the productivity of a dragline can be increased. Sections 4, 5 and 6 discuss the construction of the scale model dragline, the designing of the experiments and the design and construction of the test buckets and the soil selection. Section 7 discusses the results of the tests, section 8 is aimed at understanding the digging process qualitatively and section 9 presents conclusions that can be drawn from this project and recommendations on areas in which research is needed.

It was found that widening the bucket and lowering the hitch would result in better performance, but there are problems associated with such changes. Discussions should be held with different mines to see which changes will be feasible and a 6 cubic meter bucket should be built and tested to eliminate possible scaling errors.

2. SHORT OVERVIEW OF STRIP MINING

When comparing strip mining to the more costly and complex underground mining it is obvious that it plays a very important economic role. Panagiotou (1990) points out that opencast mining is easier, safer and requires less initial investment than underground mining.

2.1 Determining the lithology structure

According to Morey (1990) most identification is made by drilling holes and analysing the drill cuttings and cores in the region of interest. Drill-hole logging techniques combined with cores and drill hole cuttings are used to interpret the lithology sequence. These techniques (like the density log, calliper log, resistivity log, etc.) make use of differences like density and electrical resistance of the different materials to determine the position of the coal layers. The data is then presented on a number of different charts and maps to present and interpret data.

It is essential to pick up the coal structure accurately, including possible faults and joints (as described by Ward, 1990). Faults can render an attractive area unminable and joints are natural weaknesses in the rock structure of which advantage could be taken in the mining sequence. The structure influences the equipment selection. The amount and quality (for example calorific value) of coal is of importance in determining the feasibility of the operation as a whole.

2.2 Designing of the mine

Morey (1990) classifies the designing of the mine into three groups:

1. Geologic (depths, widths and slopes)
2. Mechanical (operating radii, reach, capacity, ground pressure and cycle time)
3. Operational (scheduled hours, operator efficiency, availability, utilisation, mobility and required coal production rate)

The above only goes to show the wide range of variables that need to be taken into account when designing a mine and each of the groups has to be dealt with in detail, requiring a lot of experience, research and expertise.

2.3 Overburden removal

In an opencast mine the biggest cost is associated with removal of the overburden and therefore this is the process where efficiency is the most important (Chatterjee et al., 1975; Morey, 1990; Streck, 1981 and Kemp and Chapman, 1978).

The overburden is classified by Aiken and Gunnett (1990) into the following categories:

1. **Topsoil:** Support vegetation as must be replaced after coal extraction.
2. **Soft overburden:** Can usually be excavated without blasting and is used in recountouring of the landscape.
3. **Medium hard:** Includes a wide variety of rock and light blasting is required for excavation with a dragline.
4. **Hard rock:** Drilling and blasting must be performed.

The different methods used for stripping (removing overburden) are vast and they can be used in a number of combinations. The most common methods are Draglines, Bucket Wheel Excavators (BWE), Truck/Shovel/Front End Loader combinations, Scrapers and Hydraulic Mining. Aiken and Gunnett (1990) defines the stripping ratio or stripping index as the volume of overburden that needs to be removed per volume of coal uncovered. Therefore:

$$Stripping\ ratio = SR = \frac{Volume\ of\ overburden\ removed\ (m^3)}{unit\ volume\ of\ coal\ (1\ m^3)} \qquad (2.3.1)$$

2.3.1. Draglines

Draglines are the most popular stripping tool. In Chapter 3 this statement is shown to be valid and a comprehensive discussion on draglines as well as mining methods associated with them are presented. In the following sections other mining methods that are used are discussed and compared to the dragline mining method.

2.3.2. Bucket wheel excavators (BWE)

The bucket wheel excavator is a machine having a rotating wheel with buckets mounted on its perimeter (almost like a watermill) on a sometimes extendible boom. The bucket capacity can be up to 6.3 cubic meters (Jinarajan, 1982). The machine is mounted on crawlers, and can rotate around a vertical axis.

The two biggest advantages of the BWE are its enormous production rate and its high breakout force when compared to draglines. Strech (1981) points out that in deep mines (50 to 150 m) the high production rate of the BWE is needed. Other advantages of the BWE is that it is generally used on wide operating benches (resulting in more stable pit slopes), reclamation is easier (Aiken and Gunnett, 1990) and manpower requirements are low (Jinarajan, 1982). The excavated material is preferably delivered to a belt conveyer (rail wagons are also an option) (Jinarajan, 1982), which eliminates the need for mobile transport and saves the associated high fuel costs (Aiken and Gunnett, 1990).

The BWE is mostly used in soft overburden and, combined with the higher breakout force, the need for blasting is eliminated (Jinarajan, 1982 and Morey, 1990), although blasting is sometimes used in conjunction with BWE (Atkinson et al., 1986). Improvements in the design of BWE over the last few years allows the handling of harder digging conditions (Aiken and Gunnett, 1990). The facts that it is not applicable in hard digging conditions, that it is less flexible than a dragline for example and that (being a continuous excavator) if one element of the system breaks down, the whole system is at a standstill, are the biggest disadvantages (Aiken and Gunnett, 1990).

In all operating methods two cutting techniques are used (Morey, 1990) - drop cutting (cutting vertically down) and terrace cutting (cutting horizontally across the face). Aiken and Gunnett (1990) show three operating methods (with all three methods employing terrace cutting) - block excavation, bench excavation and lateral block excavation. The reason for employing terrace cutting is that it assists in the reclamation process as discussed in section 2.5. Atkinson et al. (1986) points out that the most important operating characteristics is digging height, cut width and boom length.

Morey (1990) mentions an interesting development in increasing the productivity of the BWE that might have application in other operations. At one location plugging was a problem - it was overcome by a loosely fitted chain in a bucket back that had been cut out. By doing so the bucket tended to be self cleaning.

2.3.3. Shovel/Truck/Front End Loader combinations

According to Aiken and Gunnett (1990) shovels can be grouped under stripping or quarry shovels, the first being used for overburden removal and the latter for loading onto trucks or rail cars.

Stripping shovels: Shovels can be used as the primary means of overburden stripping or it can be used in conjunction with draglines or BWE. Their advantages are that they have a higher breakout force than the dragline and can also handle material having a lesser degree of fragmentation (Jinarajan, 1982). It is however less versatile than the dragline because of the higher ground bearing pressure, which prevents them from being positioned near an edge or on spoil piles (see the extended bench mining method used with draglines that is described in section 3.1.1.2) (Jinarajan, 1982). According to Morey (1990) the operating cost of shovels is higher than that of draglines and Sargent (1990) points out that the last stripping shovel was sold in 1969 and although there are still some in operation, they will eventually phase out. The largest stripping shovel has a bucket capacity of 180 cubic yards and a 215 ft boom (Jinarajan, 1982 and Morey, 1990).

Shovel/Truck: Shovels can also be used to load excavated material onto trucks for transport (quarry shovels). Truck loading shovels can be of two types - hydraulic or rope shovels. Hydraulic shovels have a maximum bucket capacity of about 30 cubic metres. Rope shovels are either electrically or diesel powered. Electric rope shovels are relatively cheap (but not very mobile), are very reliable and has a long life (Adams, 1990). Diesel rope shovels are more mobile, but have the disadvantage that it loses power at high altitudes (Jinarajan, 1982). Adams (1990) claims that hydraulic diesel shovels have gained a lot of ground in the last decade and are challenging draglines and rope shovels. While the lighter hydraulic shovel has a shorter life than the rope shovel, the lower ownership cost and higher productivity make it a lower cost machine (Jinarajan, 1982).

The shovel loads accurately onto the trucks and have a short cycle time. It should be positioned in such a way that it can spot trucks on both sides. The shovel dumping height must be higher than the truck loading height and the load should not be swung over the truck cabin (Jinarajan, 1982).

The main advantage of this system is its versatility (Jinarajan, 1982 and Bertoldi, 1977). Another advantage is that if one of the units breaks down the other still goes on operating, whereas in the case of the BWE or dragline the whole stripping operation comes to a standstill. Disadvantages of the system are the dependence on manpower and the high fuel costs associated with it, as well as the fact that it becomes less productive as depth increases (Jinarajan, 1982). It is also limited by steep grades, a large amount of road construction is needed and dust generation is high. A shovel/rail transport system can save on fuel but it is not as mobile and the shovel is unproductive during train trips (Jinarajan, 1990). According to Karpuz (1990) shovel productivity can be improved by up to 15% when assisted by blasting in medium digging conditions. According to Morey (1990) overburden removal by shovel/truck costs three times as much as dragline stripping, but it is easy to match the system to production goals - simply increase the fleet.

Trucks are either of the rear dump or bottom dump type. The bottom dump type is used when the material is unloaded over a hopper or is spread and the rear dump type is used to dump over an edge (Jinarajan, 1982). The excavator should fill the truck in 3 to 5 passes (Aiken and Gunnett, 1990 and Sargent, 1990). The biggest trucks in use are 119.8 cubic metres, designed to carry 176 tons of material.

Front end loader/Truck: Using front end loaders to load trucks instead of shovels provides more mobility and flexibility, but the loading operation takes longer and the maintenance cost is higher (Jinarajan, 1982). Over short distances the front end loader can also be used in a load and haul type operation (Adams, 1990)

2.3.4. Scrapers

Scrapers as the primary means (often used in conjunction with other methods) of overburden removal should only be considered in soft overburden with depths of less than 20 meters (Morey, 1990). Morey (1990), as well as Aiken and Gunnett (1990), classify scrapers into four categories:

1. **Single engine scraper:** It is usually assisted by a dozer in the loading stage and it can handle a wide range of materials. The reason for dozer assistance is that it is uneconomical to give the scraper more power than needed for haulage because of the high percentage of time spent transporting overburden, resulting in the scraper being underpowered for the loading cycle (Jinarajan, 1982).
2. **Elevating scraper:** It self loads with a powered elevator, but is limited to material of gravel size.

3. **Dual engine scraper:** It can be used in wet conditions since it has got better traction.
4. **Push-pull scraper:** It is dual engine scrapers working in tandem. The lead scraper is push loaded by the second and then the second scraper is pull loaded by the first. They then separate and haul as individual dual engine scrapers. In this way the need for a dozer is eliminated

Scrapers are often used for prestripping, since the topsoil (which is often unconsolidated material on which a dragline cannot be positioned) is then removed - this topsoil can be stored and reclamation is made easier (Jinarajan, 1982 and Morey, 1990). Prestripping with scrapers can also help in eliminating dragline rehandle (Aiken and Gunnett, 1990 and Morey, 1990). Another advantage of scrapers is that pit width does not influence the efficiency of the operation and it is very safe (Morey, 1990). Downtime on one unit also does not influence other units and the overburden removal rate can be matched to the required coal output by simply introducing other units into the system.

Disadvantages of scrapers are that their use is limited to loose formations and they are not efficient in wet conditions (Jinarajan, 1982). Road construction is high and the manpower needed is higher than with draglines. Operating and maintenance costs is high, but initial investment costs is less than for draglines. The process is limited by steep grades (Morey, 1990).

2.3.5. Hydraulic Mining

This method is limited to soft overburden and huge quantities of water is needed (Jinarajan, 1982 and Aiken and Gunnett, 1990). Jinarajan (1990) claims the method to be 20 % cheaper than dragline side casting, but points out that the application is very limited. The material is excavated and mixed by pumping high pressure water onto it and the slurry is pumped away.

2.4 Loading and conveying of coal

Light blasting is sometimes needed to break the coal layer (Woodring and Sullivan, 1990). The coal is loaded onto dumptrucks or conveyer belts. Normally the coal will be loaded with rope or hydraulic shovels onto dumptrucks, but scrapers can be used for loading and hauling of coal (Morey, 1990). The space needed for loading the coal at the bottom of the pit determines the minimum width of the strips that are to be mined.

In order to maintain a balance in the production cycle, the coal removal equipment must be matched to the overburden removal equipment, but, since the overburden removal is the critical factor as far as time and costs is concerned, the coal removal equipment will generally have excess capacity (Morey, 1990). This will be of good use when a dragline has finished a strip and has to wait for coal removal before it can continue in the opposite direction on the next strip (see section 3.1.1.1 where deadheading of a dragline is discussed).

When thin partings are found between the coal layers, the coal loading equipment can be used to remove it as draglines for example are not used on thin partings (Jinarajan, 1982).

2.5. Reclamation

After the coal has been extracted the landscape has to be returned more or less to its original shape, making sure that any toxic material is buried. The topsoil has to be in place, otherwise vegetation growth will not be supported, and the area has to be seeded. If the above measures are not taken the mine could face severe financial penalties.

The mineral content of the spoil piles has to be well known, otherwise run-off water can cause huge damage to the ecology (Aiken and Gunnett, 1990). Removing the overburden can take place in horizontal and vertical layers. In order to store the topsoil and bury toxic waste, stripping horizontal layers is preferable (Bandopadhyay and Ramani).

3. DRAGLINES AND THEIR OPERATION

A dragline is an overburden stripping machine that combines the excavation and transportation of overburden into one motion, which is one of its biggest advantages. It is used to mine shallow coal deposits and is generally assisted by blasting. It has a boom extending at an angle of about 38 degrees to the horizontal. From the boompoint down and from the machine outwards, cables - that are wounded onto drums on the machine - run to a bucket that is manipulated by changing the lengths of the cables. The machine can rotate on its base around a vertical axis. It scoops a bucket full of overburden, swings to the side (usually through about 90 degrees) and the material is dumped where the coal has already been removed. In so doing the area is mined in narrow strips (between 25 and 60 meters wide), starting where the stripping index is the lowest. The first cut that is made is termed the box cut. Each strip is divided into a number of blocks (the strip width, overburden height and block length are the important parameters) which are then excavated from several dragline positions (Humphrey, 1990; Francis, 1995; Jinarajan, 1982; and Morey, 1990). Figure 1 shows a typical dragline and its components.

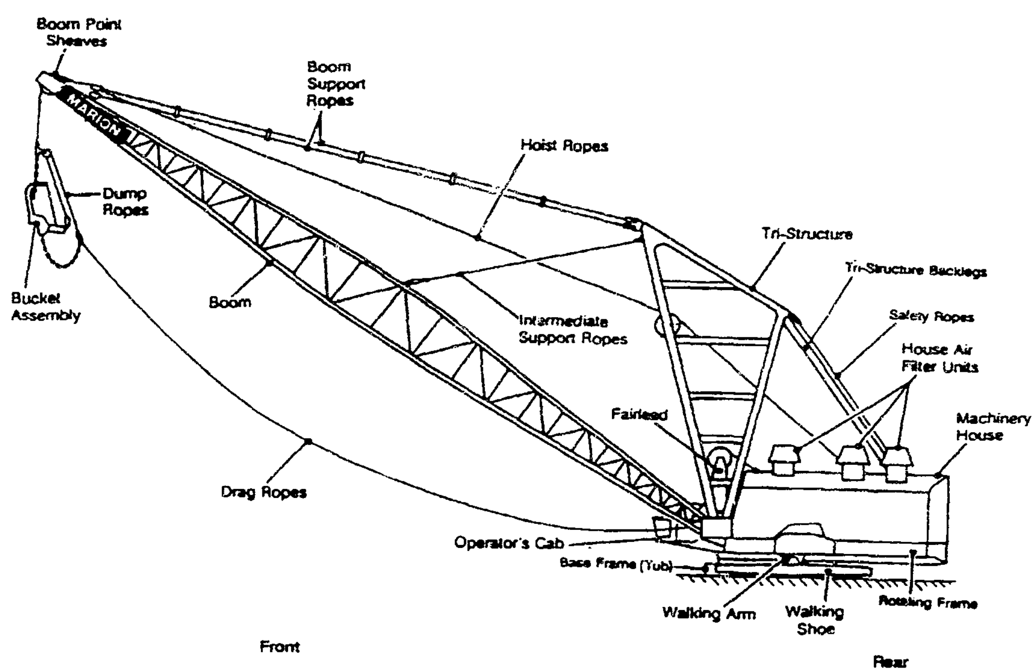

Figure 1: A dragline with its components

A dragline bucket along with its terminology and rigging (chains and cables attached to it) are shown in Figure 2:

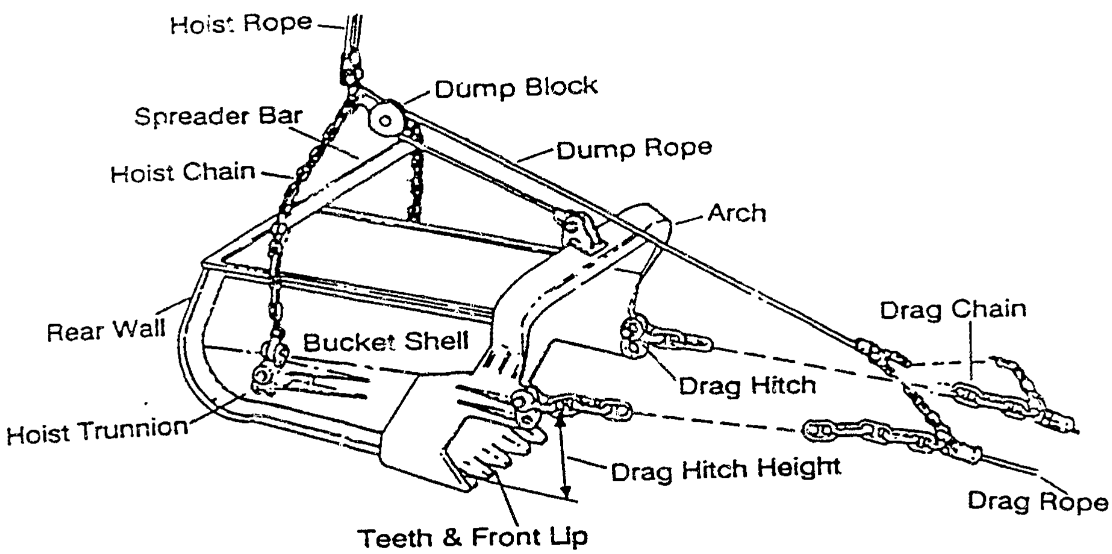

Figure 2: A dragline bucket and its rigging.

It can be seen from the figure that the drag hitch is the connection point for the drag chain and the hoist trunnion the connection point for the hoist chain. The hitch height is defined as the vertical distance from the bucket floor to the hitch and has a very big influence on filling performance. The angle of attack of the teeth is defined as the angle between the bucket floor (horizontal) and the forward face of the teeth. The drag, hoist and dump ropes are also shown. Figure 3 shows the carry angle as will be used in subsequent discussions - it is the angle between the bucket floor and the horisontal. The drag angle is defined in Figure 20 in section 8 - it is the angle the drag ropes make with the horizontal.

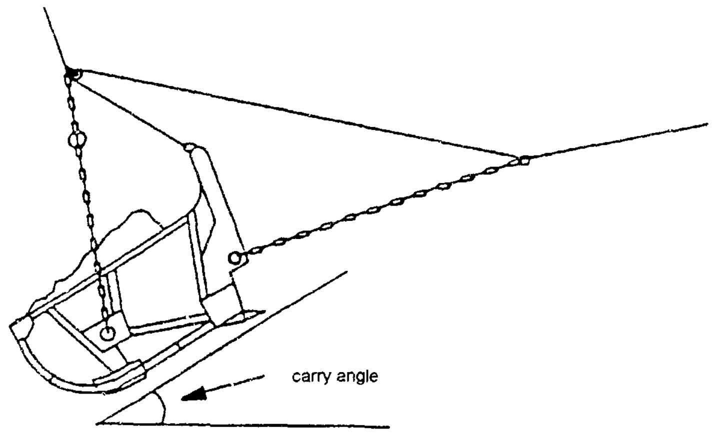

Figure 3: The dragline bucket carry angle

Draglines can be grouped into three classes (Aiken and Gunnett, 1990):

1. **Truck mounted draglines:** These are not for overburden removal and will not be discussed here.
2. **Crawler mounted draglines:** Bucket capacity is less than 19 cubic meters and it is usually diesel powered for mobility (Humphrey, 1990).
3. **Walking dragline:** These are the most common draglines and the rest of the discussion will deal with them. The machine rests on a big round tub (because of its weight) and walks with a crankshaft mechanism which puts the shoes down, tip the machine forward and then, with about 80% of the weight of the machine on the shoes and 20% on the front of the tub, the machine is slided backwards in steps of about two meters. The largest bucket fitted to a machine is 168 cubic meters (although the highest percentage of buckets falls in the 45 to 65 cubic meter class) and the longest boom is 128 meters (Humphrey, 1990 and Jinarajan, 1982). As Rowlands (1991) noticed, the size of draglines have not increased in the past few years - other ways to increase productivity has been developed. Recently however the largest dragline ever built, the P&H 9160, was offered on tender. The machine weighs about 7700 tons. Humphrey (1990) mentions that some of the smaller draglines have live booms (meaning the angle of the boom can vary). This

obviously has big advantages in terms of having a long reach at first and a higher dumping height (and shorter reach) later (see figure 4 for definitions).

The key parameters for a dragline are the boom length (and angle) and the bucket capacity. The boom length and angle determine the operating parameters of the dragline, being dumping radius, dumping height and digging depth, and it also influences the size of the bucket that can be rigged to it (Jinarajan, 1982). Figure 4 shows a dragline positioned in a pit with the appropriate definitions of the operating parameters. R*d* is the dragline dumping radius, R*e* is the effective reach and S*o* is the stand-off distance. H*o* is the depth of the overburden or the bench height and W*o1* is the pit width. The dumping height is not shown, that is the maximum height at which the dragline can dump, measured vertically from the level on which the dragline stands.

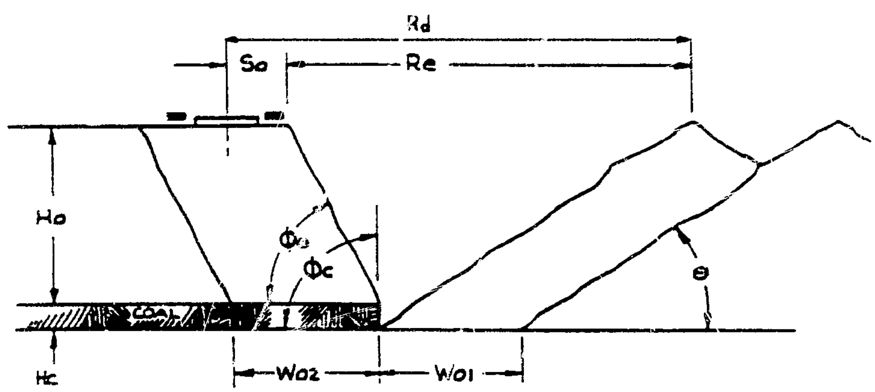

Figure 4: Dragline operating parameters

The dragline cycle consists of the following steps (Rowlands, 1991 and Humphrey, 1990):

1. **Bucket spotting:** This is the positioning of the bucket to start filling it.
2. **Bucket filling:** This is the most complex and the least understood part of the dragline cycle. It can account for 25% of the cycle time and therefore optimisation is very important.
3. **Disengage, hoist and swing:** When the bucket is visually full the operator disengages it by applying tension to the hoist rope. While swinging, the drag rope is paid out (for

dumping occcurs under boom point) and the hoist rope is pulled in (to obtain an adequate dumping height).

4. **Dumping:** Under boom point the tension in the drag rope drops, resulting in a drop in the tension of the dump rope. This causes the bucket to tilt forward and the material is dumped onto the spoil pile.
5. **Return swing:** The bucket is swung back, the hoist rope is lowered and the drag rope is pulled in to position the bucket for the next cycle.

The dragline offers a lot of advantages over other mining methods. It is widely regarded as the best combination between productivity and versatility - making them the most popular stripping tool (Bandopadhyay and Ramani; Hrebar et al., 1967 and Adams, 1990). Although a BWE can move a cubic meter of overburden at a lower cost than a dragline, its application is very limited (Bertoldi, 1977). Bertoldi (1977) also performed a cost analysis of the productivity of mining a hypothetical deposit by means of a dragline, truck/shovel and scraper mining system. Including depreciation, insurance, etc., the dragline was found to be the most efficient, in other words, the dragline had the lowest overall cost per cubic yard of overburden moved. It can also handle various types of material, have low operator fatigue, can dig above and below bench level and has a low bearing pressure (meaning it can be situated close to edges) (Humphrey, 1990; Aiken and Gunnett, 1990 and Steidle, 1979).

The biggest disadvantage of the dragline is its high initial cost. With draglines being the most popular stripping tool and with stripping accounting for the highest cost on the mine, the overall success of many strip coal mines depends primarily on the efficient use of draglines to remove overburden (Chatterjee et al., 1975). Other disadvantages are the fact that it has got a relative low breakout force, the distance that the material can be transported is small, a dozer is usually needed in support and fragmentation can influence the productivity to a high degree (Bertoldi, 1977).

The coincidental point for a dragline is defined by Humphrey (1990) as the point where the hoist, swing and drag times are equal and as short as possible for dumping - this point obviously changes with the pick-up position of the bucket and the bucket rigging. If only the hoist and swing are considered (at their maximum speed for the given dragline) the bucket follows the swing - hoist coincidental curve. If dumping occurs above this curve, the cycle is hoist dependant (or hoist critical) - meaning the swing must wait on the hoist. If dumping occurs below this curve, the cycle is swing dependant (or swing critical) - meaning the hoist must wait on the swing (Humphrey, 1990 and Morey, 1990).

Improvements in dragline design that have received attention, apart from having bigger, more powerful machines, have been in the development of aluminium booms. The booms, being lighter for the same strength, can be longer, swing time is reduced and buckets can be bigger. The tubular members are filled with gas and pressure sensors warn against the development of cracks (Adams, 1990). There are also machines working with triangular booms, but Steidle (1979) found them to be inferior to rectangular booms. The reason for this is simply that the loading imposed on the boom during the swing cycle is shared between the two outer members (the third member being on the neutral axis for swing loading) and this results in a more uneven load distribution than is the case with a rectangular boom.

3.1. Mining methods

The most common mining methods associated with draglines will be discussed subsequently. These form the basis of all operations and can be used in combination with shovels, shovels/trucks/front end loader combinations, scrapers and BWE, depending on the lithology of the mine. The number of individual systems that can be used in total is therefore vast. The different mining methods will be discussed under single and double seam applications.

3.1.1 Single seam methods

3.1.1.1. Side cast

This is the simplest and most desirable mining method. The overburden is removed in thin strips from the area that is mined. Each strip is divided into lengths (the cut, set or block length), which is excavated from several tub positions. The dragline swings through about 90 degrees and casts into the empty pit where the coal has already been removed (Humphrey, 1990). Figure 5 shows the dragline position with respect to the pit when performing side casting.

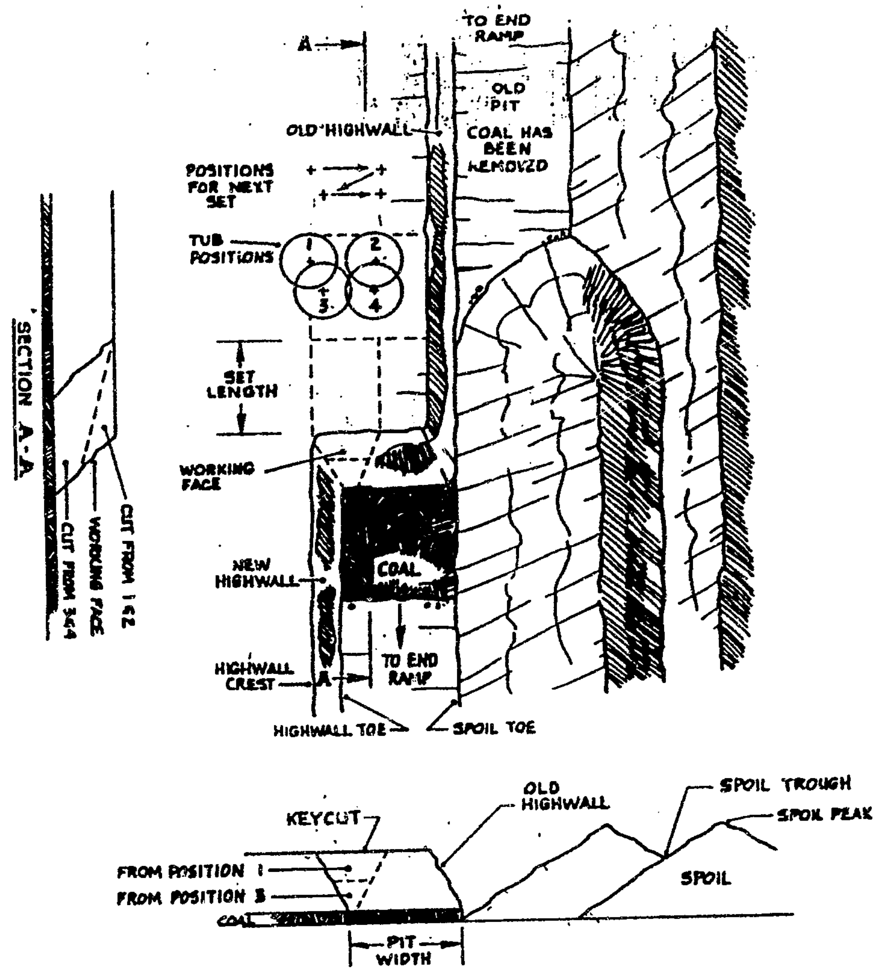

Figure 5: Simple side casting

The block is excavated with the dragline in several positions as shown in Figure 5. From position 1 the key cut (the cut fixes the alignment and slope of the new highwall) is made. According to Morey (1990) the key cut has to be made since the dragline cannot control the bucket against an open face, which would result in the width of the strip (or cut width) getting less as the dragline progresses. If the coal deposit is shallow positions 1 and 2 might be adequate to mine the block, but with deeper overburden the drag ropes will be pulled into the face and the dragline will have to be moved forward, utilising positions 3 and 4. Positions 2 and 4 should be as close to the previous highwall as possible (this is where the low bearing pressure of the dragline becomes an advantage) as this will maximise the reach of the dragline. The method as described here requires no rehandle, which, along with its simplicity, are the main advantages.

This paragraph is applicable to all of the mining methods and not just to the simple side cast method. When the dragline reaches the end of the strip, excavation cannot continue immediately since the coal in the final set must be removed first. The dragline can either walk back along the strip (termed deadheading) and start excavating the next strip in the same direction, or it can wait for the coal to be removed and start excavating the next strip in the opposite direction (termed laying over). Deadheading could be a problem though if the soil top layer is not firm enough to support the dragline weight and the choice obviously must depend on the strip length. Some mines opt for deadheading if laying over would result in more than two shifts being lost (Morey, 1990). If laying over is the selected option, maintenance should be scheduled for that period (Humphrey, 1990).

3.1.1.2. Extended bench

Figure 6 shows the extended bench method.

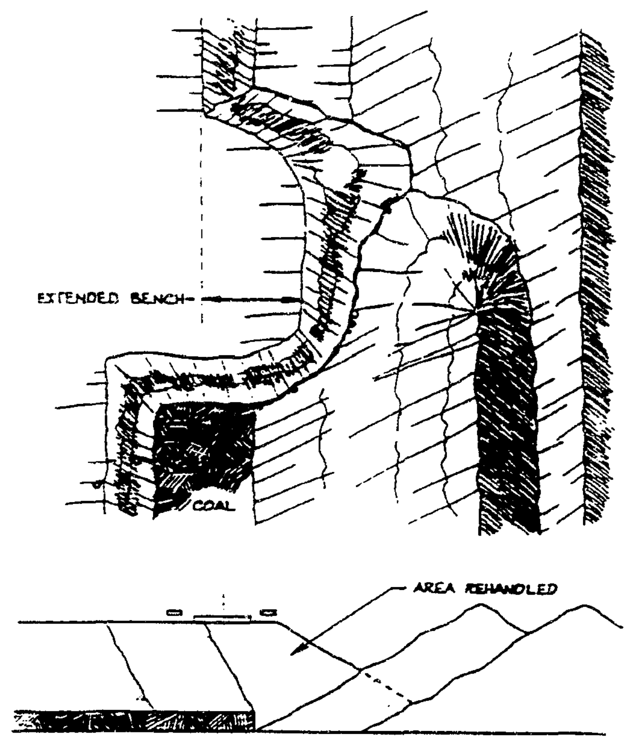

Figure 6: The extended bench mining method.

This method is used when a mine has to make best use of the existing equipment and the reach of the dragline needs to be extended. The dragline support level is extended cut from the old highwall closer to the spoil. The material from which the extended bench is built will normally come from the key cut. The material is then levelled and the base prepared for the dragline. Here the lower bearing pressure is of advantage again as this method cannot be employed with stripping shovels. The dragline is then repositioned on the extended bench to remove the material from the final cut and the previous extended bench. The disadvantage of this method is that some material (the material used to form the extended bench, coming from the key cut) must be rehandled and in doing so lowers the productivity of the dragline. (Aiken and Gunnett, 1990 and Humphrey, 1990)

3.1.1.3 Advance bench

Figure 7 shows the advance bench method.

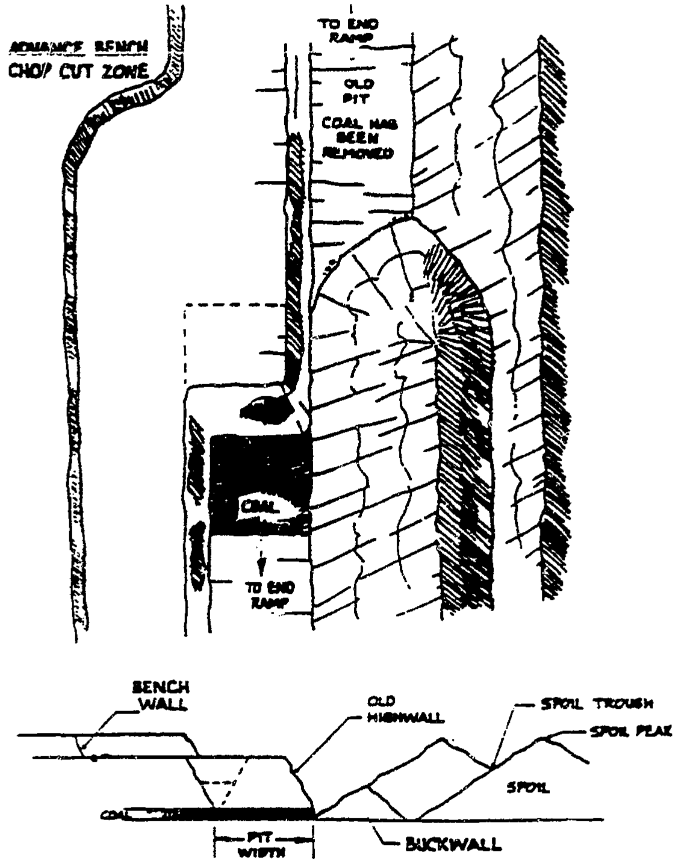

Figure 7: The advance bench mining method.

This method is used when the terrain is uneven or the top layer is unconsolidated material. The overburden is divided into an upper and lower bench. The dragline is positioned on the lower bench and the upper bench is removed ahead of the dragline by means of chop cutting (a digging method in which the bucket is dropped teeth first into the overburden that is above tub height - this is a very wear intensive digging method and the productivity is low because of low fill factors, long swing angles and longer filling distances). Sometimes a buckwall at the toe of the spoil pile will be built from more competent material first to contain the unconsolidated material.

The advantages of this digging method are that the required dragline reach is shortened, rehandle may be avoided, a level return path for deadheading can be provided and the dragline is positioned on more stable material. The disadvantages are that, during chop cutting, the productivity is low and the wear on bucket and rigging is high, but the chop cutting should account for only a small percentage of the time spent digging. (Humphrey, 1990; Morey, 1990; Bertoldi, 1977; Pundari, 1981; Jinarajan, 1982 and Aiken and Gunnett, 1990). Aiken and Gunnett (1990) also mention that it might be more productive to do the prestripping (remove upper bench) by some other means, for example scrapers.

3.1.2 Two seam methods

These are mostly a combination of the above methods and will be discussed only shortly. These methods could also be used in very deep single seem applications. It is a brief summary of Humphrey (1990).

3.1.2.1 Two - bench

Figure 8 shows the two bench method.

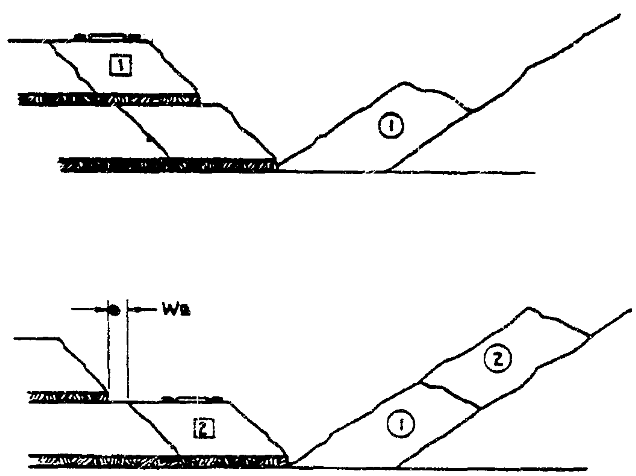

Figure 8: The two bench mining method.

This is the most straight forward of the two seam methods. The upper bench is removed and spoiled in the bottom of the previous pit. A second dragline or a second pass with the same dragline removes the lower bench and it is spoiled on top of the first.

The disadvantages of this method are that the upper burden highwall must be set back far enough to allow swinging clearance for the dragline on the lower bench. When only one dragline is being used this method also results in conflicting requirements, since a long reach is required when the dragline is positioned on the upper bench and a high spoiling height is required when operating on the lower bench. This results in the dragline being a mismatch for one of the situations.

3.1.2.2. Pullback

Figure 9 shows the pullback method.

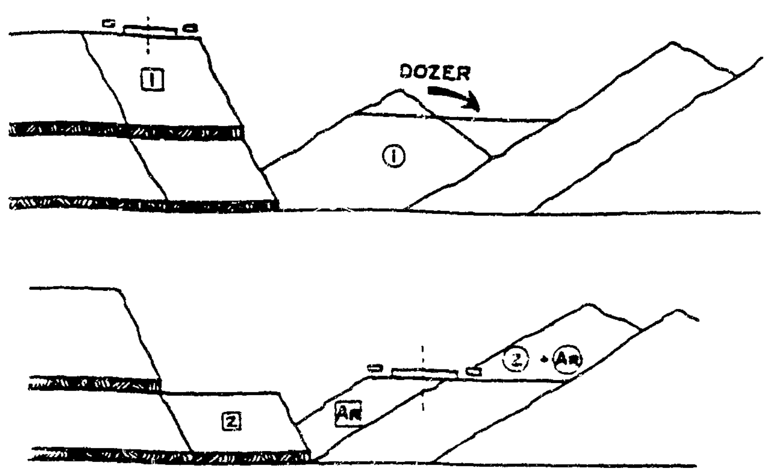

Figure 9: The pullback bench mining method.

The spoil from the upper bench is removed as in the two - bench method. This spoil is then levelled and a second pass is made with the dragline on the spoil bench. A disadvantage of this method is that the operation from the spoil bench is not as efficient as normal dragline operation.

3.1.2.3. Extended lower bench

Figure 10 shows the extended lower bench method.

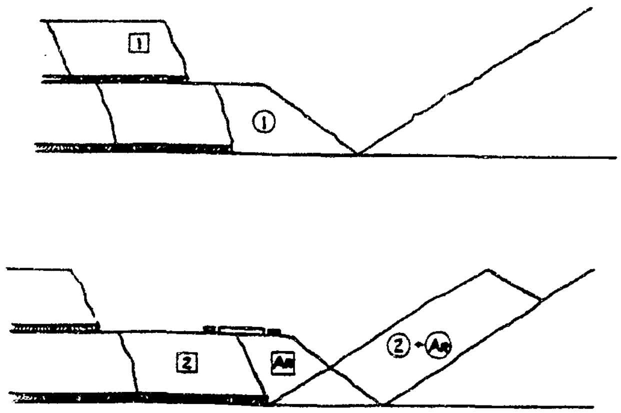

Figure 10: The extended lower bench mining method.

This method is used to help equalise the size requirements for the two benches. The material from the upper bench is placed against the lower bench highwall to form an extended bench for excavation of the lower bench.

3.2. Drilling and Blasting

This is a very important part of the excavation process as this determines the fragmentation which is essential for good productivity (Huddart and Runge, 1979; Jinarajan, 1982 and Howarth et al., 1987) and the blasting design must be such that the best advantage is obtained from the explosives energy.

Bauer and Crosby (1990B) divides drills into three groups, percussion, rotary and jet piercing drills. Percussion drills plays only a minor role, rotary drills are the most popular and can be used for drilling vertical and inclined holes while jet piercing drills is not popular due to the cost of oxygen and fuel (being a burner). Jinarajan (1982) mentions the use of rotary - percussive drills in very hard rock. According to Jinarajan (1982) the drills are either crawler or wheel mounted. Crawler mounted drills withstand the rough conditions better and is generally bigger than 250 mm, while wheel mounted drills are used for drills smaller than 250 mm and is more mobile.

3.2.1. Drilling and Blasting design

Holes are normally drilled vertically because it is easier, even though inclined holes result in less vibration, more stable slopes and better fragmentation (Jinarajan, 1982 and Ball, 1988).

Bauer and Crosby (1990), Ball (1988) and Ash (1990) consider the hole diameter, hole length, inclination, drilling pattern, the type, quantity and utilisation and the firing sequence to be the important factors in designing a blast. Figure 11 shows the blast design parameters for a vertical hole.

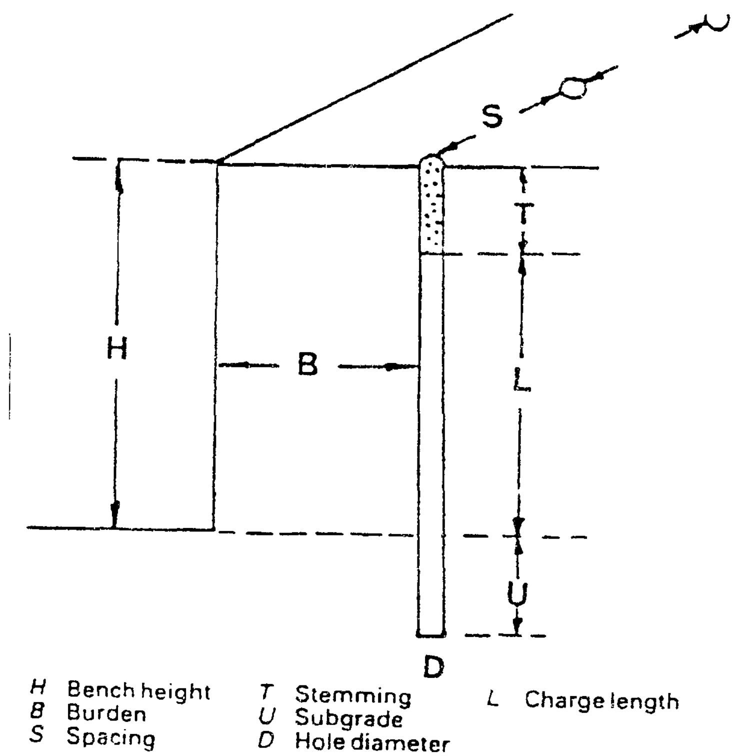

Figure 11: Blast design for a vertical hole.

According to Ash (1990) the most important design parameter is the hole diameter as that controls the explosive quantity. Ash (1990) presents empirical relationships to determine the appropriate dimensions given the bench height. Ball (1988) provides rules of thumb for the blasting design. The burden (B) should be 30 (in hard rock) to 40 (in soft rock) times the hole diameter (D). The spacing (S) should be 1 (in hard rock) to 1.25 (in soft rock) times the burden (B). The subgrade (or subdrilling) (U) should be 0.2 to 0.3 times the burden (B) except when blasting on the coal bed in which case it should be negative. According to Ash (1990) the stemming (T) should be 2/3 of the burden (B). The most common hole sizes are 143 and 162 mm (Ball, 1988). Ash (1990) and Ball (1988) agrees on the fact that the best stemming material is angular chippings. Holes can be collar or bottom primed depending on the blasting design but bottom primed blasts are generally safer and the explosive energy utilisation is better.

A ratio used to review blast results is the blasting ratio:

$$\text{Blasting ratio} = \frac{\text{volume of rock broken } \left(m^3\right)}{\text{mass of explosive used } (kg)} = \frac{1}{\text{powder factor}} \quad (3.2.1.1.)$$

Throw or cast blasting is an attempt to shift some of the overburden towards the spoil pile using the explosives energy. Morey (1990) doubts whether there is any real advantage to be gained by that, since a lot of grading is needed before the dragline can start working and sometimes the dragline is used to rebuild its own working bench.

3.2.2. Explosives

According to Ash (1990) the most important properties of explosives are density, sensitivity, sensitiveness, reaction velocity, water resistance and the fumes they produce. Since drilling is expensive, explosives with a high density are preferred as they have a higher energy output per unit volume. Sensitivity is the ease of initiation and sensitiveness the ability of the reaction to continue once started. The reaction velocity is the speed at which the explosive's reaction propagate. Water resistance is important since water can cause lower energy release values or misfire in some explosives. The fumes associated with explosives is important since it could be toxic and it could be trapped in pit bottoms if its density is higher than that of air. A good primer (identified by cap sensitivity SC ·· nore than 1.2, a high reaction velocity and immunity to the environment) is important. Holes are normally either collar or bottom primed.

Explosives that are commonly used can be divided into the following groups:

1. **OCG:** Opencast gelignite (Jinarajan, 1982). It is expensive but works under all conditions including watery conditions. Being solid cartridges however some of the explosive energy is lost in the airgap between the cartridge and the wall of the borehole.
2. **Slurries:** (Jinarajan, 1982 and Bauer and Crosby, 1990). TNT and LOX are the most popular, they are cheap and work under watery conditions. SMS (site mixed slurries) is also used.
3. **ANFO:** (Jinarajan, 1982 and Bauer and Crosby, 1990). Ammonium Nitrate Fuel Oil (ANFO) is cheap, but can only be used in dry conditions. When brown fumes are present at blast it indicates an insufficient amount of explosive or watery conditions. It has also got a low density. The low bulk density can be rectified by adding emulsions (45 to 50% of mix) resulting in heavy ANFO that can also be used in wet conditions but is more expensive. Aluminium is added to the emulsion to increase the SG and the sensitivity and glass microballoons can also be added to increase sensitivity.

3.3. Selection and sizing of draglines

The selection of a dragline is mainly based on determining the required reach (horizontal distance from tub centre to boom point) and the bucket capacity or rated suspended load based on an annual coal requirement (Hrebar and Dagdelen, 1976; Mooney and Gibson, 1979 and Humphrey, 1990). Speake et al. (1977) derived equations to determine the reach for a dragline in a single, two and three seam mining application, using simple trigonometry and considering a two dimensional cross sectional area of the pit.

The effective radius of the dragline is the dragline reach (or dumping radius) from which the stand-off (horizontal distance from tub centre to old highwall crest) has been subtracted. The stand-off varies for different applications, but should not be less than 75% of the tub diameter.

The allowable load (rated suspended load) is the weight of the bucket, rigging and overburden carried and should be selected based on a 100% fill factor even if it is less, since overloading of the boom can result in very expensive maintenance to be done later as well as downtime resulting in loss of production.

Woodring and Sullivan (1990) discussed the selection of a dragline. From three options that could be chosen they showed that in some instances a longer boom combined with a smaller bucket could be more efficient than a shorter boom with a bigger bucket. In section 3.4.2 a number of models attempting to optimise dragline operations are presented and it can be seen that to obtain a true optimum would be very difficult. Mooney and Gibson (1979) mentioned that, when selecting a dragline and trying to optimise the operation, it is common to assume a pit width (strip width), but that dragline operations is dependant on pit width, with the result that the best of the few options considered is taken which is not necessarily the optimum.

3.4. Performance and Productivity

According to Bertoldi (1977) the main factors affecting dragline productivity are deadheading (walking the machine without excavating), rehandling (moving the same overburden twice), chop down (excavating above tub level, letting the bucket drop teeth first into the overburden which is a very wear intensive and inefficient excavation method) and keycutting (excavation of a trench to form the new highwall).

Operators have a big influence on the productivity of the dragline as well as on the amount of maintenance that will be needed. This is evidenced by the large number of training facilities that exist.

3.4.1. Defining productivity

The most important performance variable for a dragline is the amount of overburden it moves in a period of time. This is dependant on the number of cycles in a given time and the volume moved with each cycle.

Number of cycles: Humphrey (1990) gives a breakdown to define the availability and utilisation of a dragline. The calendar hours (Hc) is the total number of hours in a certain period, say a year. The scheduled hours (Hs) is the time it can be expected to operate and is obtained after time for scheduled shutdowns, power shutdowns and strikes have been subtracted from the calendar hours. The available hours (Ha) is the time the machine is mechanically and electrically ready to operate and is obtained after time for repair maintenance has been subtracted from the scheduled hours. The operating hours (Hu) is the time in which the dragline is operating at full potential and is obtained after time spent on walking, bench preparation and clean-up have been subtracted from the available hours. The availability (A) is then defined as $\frac{H_a}{H_s}$ and the utilisation (U) as $\frac{H_u}{H_a}$. With the average cycle time (Tc in seconds) known the cycles per operating year can be calculated:

$$cycles\ per\ year = \frac{3600}{Tc} \times \frac{Hs}{year} \times A \times U \qquad (3.4.1.1)$$

Volume moved per cycle: When the material is blasted it expands (or swells) and the swell factor (Fs) is defined as $\frac{bank\ cubic\ yard}{loose\ cubic\ yard}$. With the bucket capacity (Bc) known and assuming a fill factor (Ff) for the bucket (defined as the percentage of the bucket that is filled) the overburden moved per cycle can be calculated as:

$$overburden\ volume\ moved\ per\ cycle = \frac{Bc}{Fs} \times Ff \qquad (3.4.1.2.)$$

From the above the total amount of overburden moved per year can be calculated by multiplying equations (3.4.1.1.) and (3.4.1.2.). To determine the prime amount of overburden moved per year the rehandle (percentage of overburden that must be handled a second time) must be taken into account. From the above it can be seen that the biggest bucket will not necessarily have the highest productivity, since a smaller, lighter bucket might have a faster cycle time.

Of secondary importance is the amount of electrical energy consumed during the cycle, with a lower energy consumption per cubic yard moved associated with a higher efficiency. Electrical power is consumed in huge quantities by the dragline and substantial financial gains is possible if reductions in this area is possible.

3.4.2. Optimisation

Optimisation of dragline excavation have received much attention. The work that has been done in this area, mostly was in one of four areas of which two will be discussed here. The one was in mathematical models calculating or optimising dragline productivity and a second was in monitoring the efficiency of the dragline so the mine could know how the dragline was performing. Then efforts were also made to increase the availability of the machine (decreasing the amount of unscheduled maintenance) and a lot of effort went into optimising drilling and blasting.

3.4.2.1. Mathematical models

Optimising the overburden removal rate is very difficult, since a lot of factors need to be taken into account. Several models in which the efficiency of mining a block or strip, with the block and dragline dimensions as inputs, have been developed. Walk and swing times are based on regression analysis done on actual recorded times and the validity of the models were compared with actual dragline operations. Averages filling times are used. Some of the models could incorporate rehandle in which case it was used for the construction of an extended bench. (Bandopadhyay and Ramani; Chatterjee et al., 1975; Huddart and Runge, 1979 and Baafi et al., 1995)

Chatterjee et al. (1975) summarises the most significant variables as:

1. Number and location of dragline positions.
2. Zones of digging from these positions.
3. Length, depth and width of each cut.
4. Length of strip.
5. Angle of repose of spoil pile.
6. Swell factor.
7. Dragline dimensions and specifications.

These variables are interrelated, for example the pit width (or cut width) is a function of the coal removal equipment (space needed), the overburden depth, the blasting pattern, the material characteristics and dragline dimensions (Humphrey, 1990). Huddart and Runge (1979) notes however, that the coal deposit and overburden depth is fixed and that the dimensions that can be varied are pit width, bench height and block length (usually the mine planning is done for an existing dragline, in other words the dragline dimensions is fixed as well). The **bench height** is defined as the height above the coal at which the dragline is positioned. The **cut length, dugout length or block length** is the length between major digout cycles. The **panel width, cut width, block width or strip width** is the width of each consecutive strip that is mined.

The following general guidelines are suggested (the suggestions made by Chironis (1978) is based on productivity monitoring and not on a simulation model):

Block length: Of the three, dragline productivity is least sensitive to block length (Huddart and Runge, 1979). According to Chironis (1978) the block length should be as long as possible. This view is supported by Morey (1990) as long as the bucket does not need to be cast beyond the reach of the boom.

Block width: Of the three, dragline productivity is most sensitive to block width (Huddart and Runge, 1979). According to Bertoldi (1977) the narrowest practical pit usually is the most economic since rehandle is minimised, dragline cycle time is reduced and reclamation is easier and cheaper because of more closely spaced spoil piles. Morey (1990) notes that narrow widths allow more flexibility with the placement of spoil (which is of advantage for road construction) and that wider pits require less dragline walk time. It is believed that a wider pit is beneficial up to the point that rehandle is required, but the space needed to load coal at the bottom of the pit, the mine layout and overburden depth may require pit widths in which rehandle is needed.

Bench height: According to Morey (1990) the bench height should be as high as possible (within the reach of the dragline). This is probably to eliminate the amount of pre-stripping that is done with machines that are less productive than draglines.

It is also suggested that the key cut should be as wide as possible without creating rehandle (Chironis, 1978) and that, if the swing angle exceeds 150 degrees, a full 360 degree swing should be made (Chironis, 1978 and Jinarajan, 1982) since the accelerating and decelerating times at the dumping position could then be saved.

According to (Chatterjee et al., 1975) complete 3D modelling would be of great assistance in mine planning. Francis (1995) attempted exactly that, although the project was not finished at the time of writing this thesis. The data used in the model was obtained from a Tritronix T9000 system. This allowed the calculation of filling time and payload as a function of dragline position and position where the bucket started to fill. Swing and walk times were based on regression analysis. By dividing the block to be mined (with the block dimensions as the input values) into several sections and considering all feasible combinations of excavation the optimum mining sequence could be calculated. Applied to simple side casting, the model predicted the ideal zig zag pattern of dragline positioning. Refinement was still needed and the model still needed to be extended to other mining methods by incorporating, for example, rehandle.

An Australian company, Earth Technology, developed a software package 3D-DIG that models all aspects of dragline operations and, with the overburden, coal, dragline dimensions, acceleration times and drag, hoist and swing speeds as inputs, calculates the efficiency of mining a specific piece of the mine. It graphically shows the mining sequence on the computer and can incorporate rehandle and other operations associated with the dragline (like prestripping with truck and shovel).

3.4.2.2. Monitoring systems

A number of monitoring systems have been developed to monitor the performance of draglines. These systems also guard against tightline situations and some of the systems give feedback to the operator in order for him to realise what the effect of changes in his operating style has on dragline performance. (Chironis, 1978; Kemp and Chapman, 1978; Kemp and Horvath, 1979; Nicholas, 1978).

The Tritronics T9000 Dragline Performance Monitor records more than 40 performance variables for the dragline on each cycle. These variables include the position of the bucket at start of fill and end of fill, swing angles, amount of overburden moved, dumping heights, walking times, etc. (Francis, 1995). With this information the mine has got all information needed for good mine planning, for evaluating operators and for comparison of, for example, the effect of different buckets on productivity, but it must be remembered that the operators and digging conditions can have an influence on the result.

3.5. Dragline Buckets

"Unfortunately bucket design is still an inexact science and there are no hard and fast rules to follow" (Pundari, 1981). Rowlands (1991) found this statement to be true and the work he did on draglines is still considered to be the only report on dragline bucket design (Swiericzuk, 1994).

It was pointed out earlier that the skill of the operator is of great importance, but according to Rowlands (1991) and from discussions with mine personnel, it is obvious that the bucket influences dragline performance to a high extent. Rowlands (1991) identify areas in the dragline operation in which improvements were made to increase productivity. These included more powerful swing motors to reduce cycle time, increases in machine reliability by performing preventative maintenance, improvements in pit design and mine planning (use of computers), monitoring of the efficiency of the operation, training of the operators, better blasting practices and the elimination of peak electrical loads. He then states that lightweight buckets were the only documented improvements in bucket design and that it was a neglected area - it seems as if the mines had something that worked and nobody bothered trying to improve it. When considering the amount of effort that has gone into dragline research in general, it is obvious that the need exists to improve bucket design, since it is widely known that small increases in productivity could yield huge extra earnings in terms of increased production. In South Africa it is generally accepted that an increase in productivity of 1% can yield increases in production worth R1 million a year per dragline. An increase of 9% for example for a mine having four draglines, results in roughly R36 million a year increase in production.

3.5.1. Rigging

The dragline bucket is manipulated by the drag and hoist ropes (Figure 2 on page 13). The hoist rope is used to raise and lower the bucket and the drag rope to move it forward and backward. The carry angle of the bucket (Figure 3 page 14) is controlled by the tension in the dump rope which is mainly influenced by the drag rope. (Humphrey, 1990).

Knights and Shanks (1992) developed a model to calculate the carry angle of the bucket in the two dimensional space under the dragline boom. The model was based on statics (observation of working buckets suggested that dynamic effects are on important when the bucket is dumping). The model allows easy visual comparison between different rigging options and calculates the forces in each component. Figure 12 (taken from Knights and Shanks (1992)) plots constant carry angle curves **for a typical rigging setup**. It can be noted that the constant carry angle curves more or less follows the trajectory of the bucket, meaning that the bucket could be kept close to its carry angle at disengagement almost to the dumping point. This of course is an aim of rigging design, since material is lost either from the mouth or the back of the bucket when the carry angle changes.

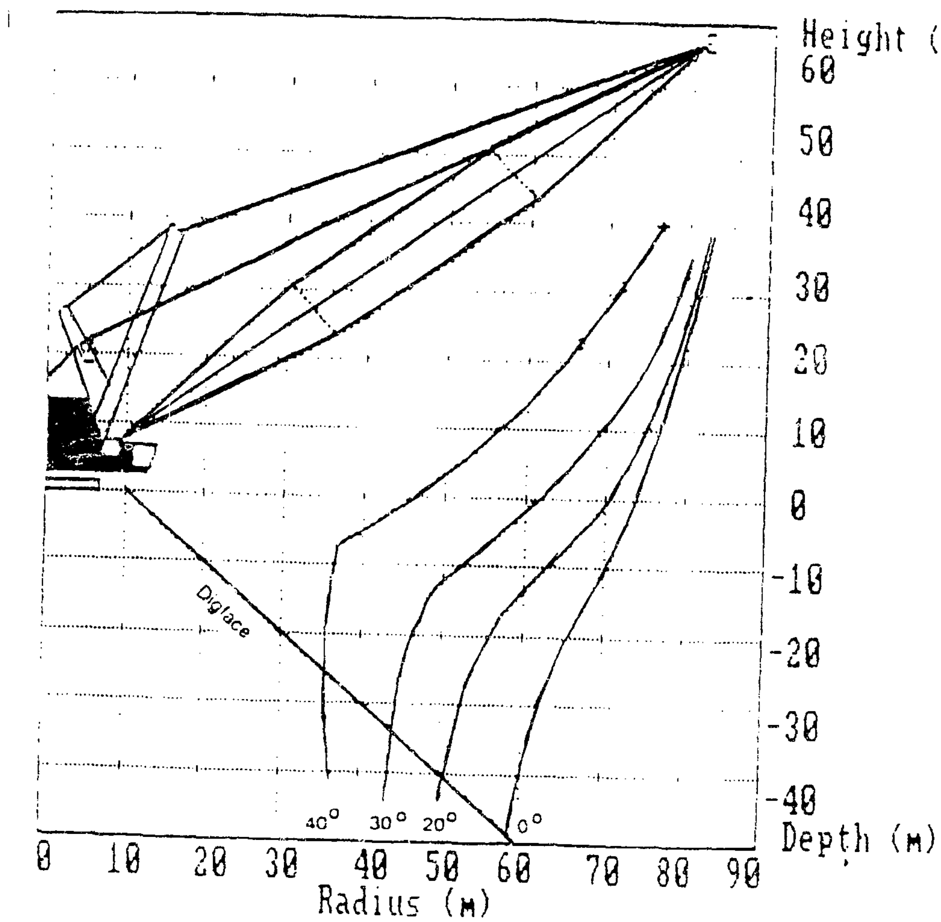

Figure 12: Lines of constant carry angle in the 2D space under the dragline boom. [Knights and Shanks, 1992]

Rigging design has changed very little since the dragline has been born. The most noticeable changes are the connecting of the dump rope to the top rail instead of the arch (only in a few cases), which is probably to reduce the loading on the arch, and the use of inside trunnions by Bucyrus Erie (BE) in order to eliminate the spreader bar. According to Ferreira (1997) the inside trunnions also suffer less abrasive wear, since the material in contact with it is at rest, while the outside trunnions is being pulled through the material continuously. During the development, that led to the inside trunnion configuration, they also discovered that the position of the hoist trunnion in particular is very important in the bucket design - shifting it to the top rail resulted in inadequate dumping, while shifting it to the back of the bucket resulted in premature dumping.

Lumley and Jensen (1996) did a rigging study on several scale size buckets and they came to the very important conclusion that, provided the bucket is set up at it's optimum carry angle by changing the dump rope length, the length of the hoist and drag chains, within reason, will not affect the productivity of the bucket to a high extent.

3.5.2. Filling behaviour

3.5.2.1. Performance measurement

The most important performance variable is the filling distance (Rowlands, 1991). Apart from the saving in time, an additional advantage of minimising filling distance is that it reduces the amount of wear on the bucket.

The other important performance variable is the amount of payload picked up by the bucket. These two variables result in the amount of overburden moved in a certain period of time which is the way a mine evaluates the performance of a bucket.

Another commonly used method of comparing the performance of ground engaging equipment is by calculating the specific digging energy (a measure of the ease of digging) (Hendricks et al., 1988). This is the amount of energy absorbed to excavate a unit volume of overburden. In the documented project, buckets of constant volume were used and were therefore compared on the basis of the amount of energy absorbed to fill.

The amount of drag power available is limited and too high drag power requirements can cause the bucket to stall. This can be overcome by applying some hoist force which causes the required drag force to drop (like a plow behind a tractor) and the bucket can commence to fill. This however will waste some time and the length to fill will be lengthened somewhat. Also, with the DC motor characteristics that reduces speed at higher torque requirements, it follows that the filling time for the same distance will be more in the case of high force requirements. All considering, geometrical changes should keep the required drag force as low as possible.

3.5.2.2. Influencing factors

Overburden: The overburden does have a big influence on the performance of the bucket, not only on the filling performance, but also on the life of the bucket due to abrasion. As was discussed in section 3.2. the fragmentation is very important. In wet conditions overburden with a high clay content can cause plugging of the bucket. The influence of overburden characteristics on dragline buckets has, apart from fragmentation, however not been well researched.

Bucket geometry: This was what was to be established in this project. The work that has been done on it previously is discussed in section 5.1. Pundari (1981) and Rowlands (1991) suggest that it would be difficult to come up with a single design that will work well under all conditions and he promotes the idea of designing buckets for specific purposes. However, one still wants a bucket that will perform well under a range of conditions. The most important factors influencing dragline bucket filling were found by Rowlands (1991) to be bucket width, hitch height, angle of attack of teeth and teeth length.

4. CONSTRUCTION OF THE SCALE MODEL DRAGLINE.

The purpose of the scale model dragline (see Figures 13 and 14) was to operate the scaled bucket in a similar fashion as a real size bucket. It is therefore obvious that the operating parts that would influence the bucket had to be designed with great care. The construction of the scale model dragline (hereafter SMD) will only be presented briefly (considering that really only the hoist and drag motors and the data aquisitioning system is necessary to obtain filling data). The author was solely responsible for designing and had one boiler maker to assist in the construction of the SMD - it amounted to more than half a year of hard work and therefore accounts for much of the effort going into the project as a whole, even though not reported in detail. There were for example a lot of general strength calculations, calculations for the expected life from bearings, etc. that are not included. The author was also solely responsible for sourcing and ordering parts which account for a lot of effort when taken down to the level of circlips, bearings and keys for shafts.

It could be argued that for the tests one really only needs a rig similar to the large scale test rig presented by Rowlands (1991), but it was realised that, once finished, the SMD had huge possibilities in terms of training and when presenting the testing facility to potential customers a scaled down version of a dragline would make a bigger impression than a rig. Another advantage of the SMD is that it can be used in productivity studies and for dynamic simulation of rigging.

4.1. General

The SMD was constructed on the tracks and rotating frame of a O&K RH6 excavator that had been scrapped. The rotating frame was broken loose from the tracks.

The swing bearing was first taken from the rotating frame, it was stripped, cleaned, lubricated and reassembled. A new mount was fabricated and used to mount the bearing on the tracks. All the cracks on the tracks and rotating frame were gouged and repaired welded. An extension to the rotating frame was made to allow for the cabin and the rotating frame was then assembled on the tracks.

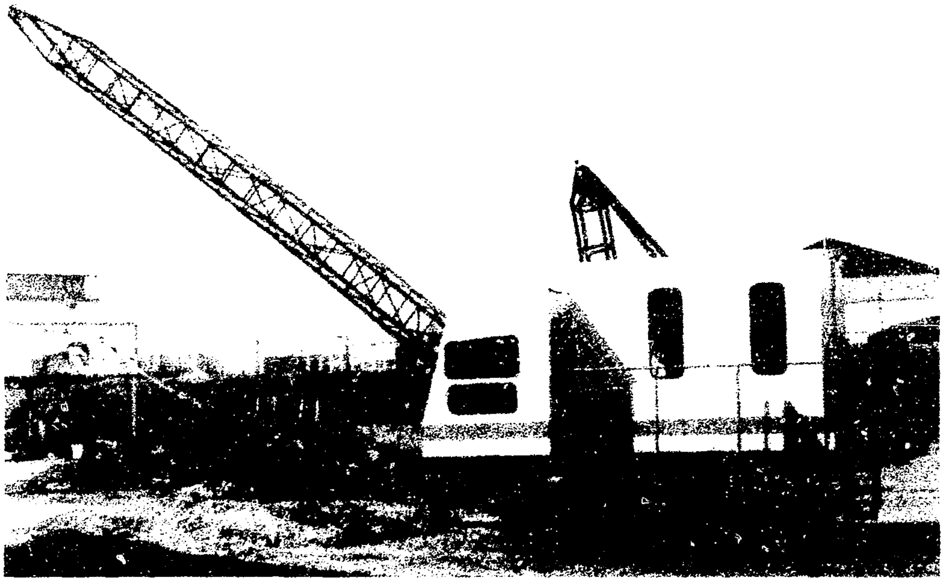

Figure 13: The scale model dragline

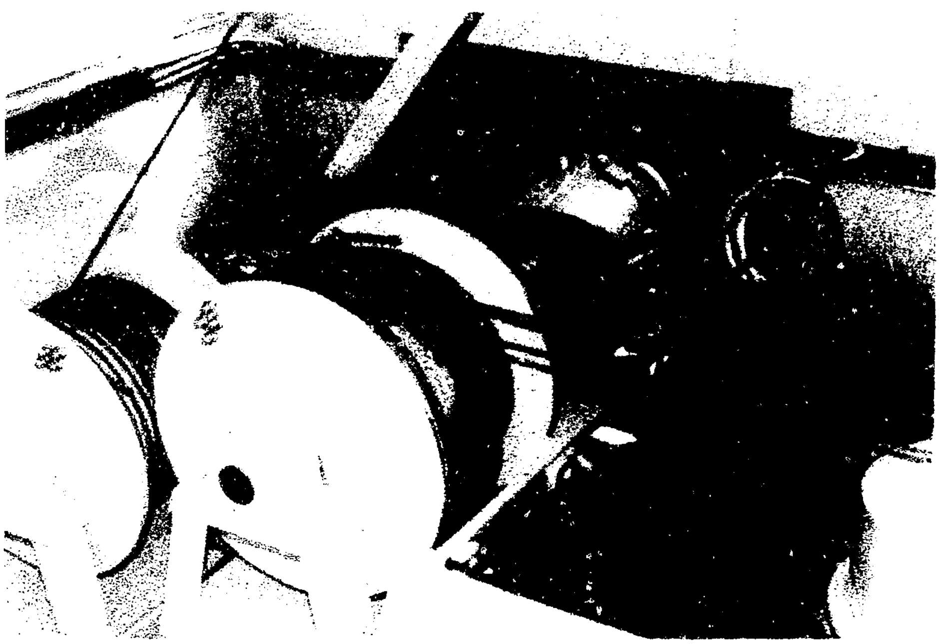

Figure 14: Layout of the swing, drag and hoist motors

The boom of the mini-dragline was made twelve metres long (scaled down 1 to 10 from a real dragline). It was made up in sections that were then bolted together. This allows the boom to be changed to virtually any length required. The support for the boom consists of the mast (from which two cables run to boom-point) which in turn is supported by the backlegs.

The hydraulic powerpack and electric motors with control panel were fitted on the rotating frame that was covered by the machinery house. The rotating frame for the machinery house was made from rectangular tubing and was covered by 3 mm steelplate. It was split in half to allow for removal of the machinery house. On the side a sliding door was fitted with windows right around. The machinery house was designed to be weatherproof - the only possible leak is were the hoist cable goes through the roof. The cabin was built in a similar way. Large windows were placed in front to allow for good visibility. The operator chair, controls and the data acquisition system were fitted inside the cabin. The cabin was also designed to be removable and was also made weatherproof.

4.2 Sizing of the Drag, Swing and Hoist motors

As far as the filling of the bucket is concerned these are the most important components by far. The same motor was selected for all three components. It was a 180 Volt DC motor. Choosing an AC motor would have been cheaper but the control would have been more expensive and, since on draglines DC motors are generally used, DC motors were chosen for the application. The control on the motors involved the ability to set a maximum speed and torque for the motors. The speed of the motors are also proportional to the amount that the controls (two joysticks for drag and hoist and two footpedals for swing) are moved. The acceleration and deceleration times can also be changed - this is important in order not to put to high stresses on the boom when the bucket is swinging and for training purposes it provides the ability to obtain the same 'feel' as on a real dragline.

The gearboxes that were chosen for the hoist and drag motors are 1:97 in-line gearboxes. The swing gearbox is a right angle bevel gear gearbox with ratio 1:105. A

worm drive gearbox could not be selected - even though the acceleration and deceleration could be changed, a power failure could cause severe damage to the gearbox (and machine) since it is not reversible and would come to an abrupt stop - this would have a catastrophic result when the machine is swinging due to the large moment of inertia. Subsequently the procedure for sizing the motors is presented.
In the end three similar motors were chosen - it means fewer different parts on the SMD and the excess power presented no problems, since it could be limited.

4.2.1. Swing motor

The power for the swing motor was determined by calculating the approximate moment of inertia around the swing axis and, wanting the dragline to accelerate from standstill to about 2.5 rpm (estimated required maximum) in 15 degrees, the torque and therefore the power requirements could be calculated. The moment of inertia was estimated to be approximately $70\,000\ kg.m^2$. This was calculated by breaking the SMD up into masses at specified distances from the rotation axis and incorporating a safety factor of 1.5 to allow for uncertainties. The angular acceleration required could then be calculated from:

$$\dot{\Theta}^2 = \dot{\Theta}_0^2 + 2\ddot{\Theta}\Theta \qquad (4.2.1.1.)$$

(Sauer et al., 1986) where

$$\dot{\Theta} = \frac{2\pi N}{60} \text{ is the angular velocity and} \qquad (4.2.1.2.)$$

N = rev/min

With $\dot{\Theta}_0 = 0$, $\Theta = \frac{15\pi}{180}$ and $N = 2.5\,rpm$ the angular acceleration was calculated as $\ddot{\Theta} = 0.131\,rad/s^2$. The required torque could then be calculated from.

$$T = I\ddot{\Theta} \qquad (4.2.1.3.)$$

giving T = 9163 Nm

With the required speed of 2.5 rpm the power required could be calculated from:

$$P = T\,\dot{\Theta} \tag{4.2.1.4.}$$

giving 2.4 kW. With the gearbox driving a 15 tooth pinion which in turn drives a 95 tooth internal gear (effectively a second gearbox) and assuming a 90% efficiency for both, the required power output from the motor is:

$$P = \frac{2.4}{0.9 \times 0.9} = 2.96\ kW$$

The above approach is conservative in the sense that the torque output from a DC motor is high at low speed and low at high speed. The calculations were done for a high torque and speed. The maximum output speed of the motor is 1750 rpm. With a 1:105 and 15:95 gearbox in series the maximum rotational speed of the SMD would be:

$$N_{SMD} = 1750 \times 1/105 \times 15/95 = 2.63\ rpm$$

The above show that the selected motor can provide the required power and that the rotational speed of the SMD is sufficient with the gearbox as selected.

4.2.2. Drag motor

The power for the drag motors could be calculated from the required force in the rope, the speed at which the bucket had to move and the radius of the drag drum. The maximum drag speed of a Marion 8050 dragline is approximately 2.5 m/s (Rowlands, 1991). Determining the maximum drag force expected, proved to be not all that easy. Rowlands (1991) presents the following empirical, linear relationship between bucket size and drag stall force by plotting data for several draglines:

$$F(kN) = 40.4 \times V(m^3) + 688 \qquad (4.2.2.1.)$$

where:

F = Drag stall force (kN)
V = Bucket capacity (cubic metres)

Obviously this cannot hold for small buckets, since a zero volume bucket would require a 688 kN drag stall force. It is however significant that the plotted data showed a linear relationship between bucket size and drag stall force. It was then argued that if data on the maximum drag force measured in similar tests for both a smaller and bigger bucket than the typical 60 litre bucket that would be used in the tests could be obtained, scaling it linear according to equation (4.2.2.2.) would result in one of the values being conservative.

$$F_{D60} = \frac{F_{DV}}{V} \times 60 \qquad (4.2.2.2.)$$

V = volume (l)
F_{D60} = drag stall force for 60 litre bucket (units same as F_{DV})
F_{DV} = drag stall force for V litre bucket.

Swiericzuk (1994) did rigging experiments on a 14.5 litre bucket for his engineering degree at the University of Queensland. The maximum drag force measured by him was about 1100 N, which translates to 4371 N (446 kg) drag force required for a 60 litre bucket, according to equation (4.2.2.2).

Rowlands (1991) measured a maximum drag force of 28300 N for a 250 litre bucket, which equates to 6792 N (692 kg) for a 60 litre bucket. From equations

(4.2.1.2.), (4.2.1.3.) and (4.2.1.4.) the output torque of the motor could be calculated as 20.19 Nm (speed of motor is 1750 rpm and power is 3700 W). With a 1:97 gearbox and assuming an efficiency of 90 percent, the output torque from the gearbox is 1763 Nm. For a 0.18 metre radius drum, the force that can be applied was calculated to be 9792 N (998 kg). It can be seen that this is sufficient for the purpose, allowing a fair margin for the uncertainties such as digging material, bucket width etc.

The maximum output speed from the gearbox is 18.04 rpm (1750/97) or 0.3 rev/s. The drag drum has a 0.18 metre radius and thus a 1.131 metre circumference and the resulting maximum speed of the bucket is 0.34 m/s, which is more than sufficient for a tenth scale model.

The above calculations show that the selected motor and gearbox easily answer to the set requirements.

4.2.3. Hoist motor

The selected motor and gearbox for hoisting the bucket are exactly the same as for the drag. The speed has already been shown to be sufficient. The force in a rope with a mass hanging to it is proportional to the mass and also to the volume since

$$m = \rho V \qquad (4.2.3.1.)$$

and therefore a similar relationship exists for the hoist force as for the drag force. Swiericzuk (1994) reports a maximum hoist force of 560 N that equates to a 2225 N (227 kg) required hoist force for a 60 litre bucket and it is clear that the selected motor satisfies the requirements.

4.3. Electrical system and controls

The electrical system is fairly simple. The incoming power source to the SMD is 380 V AC. To transfer the power to the rotating body, not twisting the cable, a slip ring was mounted on the slew distributor (see next section). From there the power was transferred to the control box that also contains the DC drives. It was then transferred to the drag-, swing-, hoist- and hydraulic motors, each having a switch in the cabin within easy reach of the operator.

The controls consist of two joysticks - the right hand side being for the drag and the left hand side for the hoist. Then there are two foot pedals for the swing and the hydraulic valves for operating the tracks. The drag joystick was wired such that the bucket moves toward the operator when being pulled and away when being pushed as on most draglines. Pulling the hoist joystick causes the bucket to lift (and visa versa) and the machine swings to the side of the foot pedal that is activated.

4.4 Hydraulics

A fairly simple hydraulic layout was chosen to drive the tracks. It consists of a gearpump capable of delivering approximately 67 l/min at 25 MPa. It is driven by a 22 kW 380 V AC motor. Calculating the actual pressure and flow rate was a problem since no specifications were available on the hydraulic motors nor on the final drive - not even from the distributors The expert knowledge of a hydraulic company (HYTEC) was called on who, from previous experience, were able to supply a system that worked. The capacity of the tank is 260 litres, which might not be enough to dissipate the generated heat in the long run, but, since the SMD would be required to neither travel often nor travel long distances, the size was limited mainly due to space problems.

The hydraulic valve also forms the controls for the tracks and was placed in the cabin within easy reach of the operator. The hydraulic oil passes through the valves (one valve for each track) and through a slew distributor that prevents the hydraulic hoses from twisting and eventually breaking. There are five lines through the slew distributor - two for each hydraulic motor (one is a supply line and the other a return line) and one for the combined leak lines. Having passed through the hydraulic motor (the direction is being determined by the valve) the oil then passes back through the slew distributor through a filter and into the tank.

As mentioned in the previous section, the incoming power is transferred to the rotating car body through a slip ring. Obviously the slip ring and the slew distributor had to be mounted on the rotation centre of the frame on the tracks - this was accomplished by manufacturing the slew distributor with a hole through its centre with the slip ring mounted on top of the slew distributor. The power cable to the slip ring could then be passed through the hole.

4.5 Rigging

It is a well known fact that the rigging of a dragline is of great importance, since the configuration influences, among others, the forces in the ropes, the carry angle of the bucket, the suspended load, the load carrying capability in the two dimensional space under the boom and the dumping characteristics.

It however proved to be very difficult to get a representative set-up. Because of competition between companies, the rigging manufacturers like ESCO and Scheffer Mechanical were not willing to supply any information regarding weights or sizes - they were concerned about the fact that Barlows might start manufacturing rigging. The wear on the rigging is also very extensive, especially on the drag chains. The chains lose mass (up to 50 percent in ten months according to Smit (1996)) and wear in the crotches cause the inside length of the shackle to increase. Both of the above factors cause the distribution of weight to vary. The rigging had to be manufactured from commercially available chain, cable and D-shackles due to time and financial limitations and therefore perfect scaling was not possible. Since this project was primarily focused on the influence of the bucket filling performance due to geometry and with a rigging testing programme already defined, it was decided to do the best possible given the limitations.

The best that could be done was to visit a mine and measure the rigging components. The rigging that was measured was standard ESCO rigging. The weights were estimated from data supplied from the mine for example for a 23 shackle drag chain with two pearlinks.

It was estimated that a drag link weighs approximately 90 - 100 kilogram and have an effective inside length of 400 mm. Scaling the effective length by a factor of ten and the mass by a factor of thousand (linear scale to the power three) a link with a inside length of 40 mm and a mass of 90 - 100 g would be representative. The best that could be done was an inside length of 39 mm and a mass of 80 g. It was slightly lighter than what was needed - but it would account for a rigging set-up having worn a bit.

The hoist chains were estimated to be between 42 - 46 kilogram and of inside length 300 mm. The closest to the scaled version that could be found was a 30 g link with inside length of 29 mm. The dimensions of the spreader bar varies according to the width of the bucket. At Optimum Colliery the spreader bar is made one meter wider than the bucket and this approach was followed in the experiments. Dimensions for the dump block were measured and scaled down linearly. The drag and hoist ropes usually are 90 mm in diameter and in a double dump configuration the dump ropes are about 50 mm in diameter. These values were scaled down linearly.

4.6 Data acquisition system

The data acquisitioning system obviously is very important. The system consists of five loadcells, an inclinometer, a potentiometer, an amplifier/power supply and a PCMCIA data acquisition card.

The loadcells were fitted in line on two drag ropes, two dump ropes (for a double dump configuration) and one hoist rope. The loadcells in the drag ropes were two 500 kg loadcells (selected on the basis of the calculations in section 4.2.1.2). The loadcell in the hoist rope was also a 500 kg loadcell (selected on the basis of the calculations made in section 4.2.1.3), while the loadcells in the dump ropes were 100 kg loadcells (based on similar calculations as the above). These were all standard loadcells, sealed against moisture, temperature compensated and boasting excellent linearity and repeatability characteristics. The sensors fitted to the bucket is shown in Figure 15. The connecting cables were tied up later to avoid interference while digging

Figure 15: Sensors fitted to bucket

The inclinometer could not be used during digging. Its usefulness comes in rigging experiments when it gives the carry angle of the bucket - after disengaging the digging material - and can be attached to the back of the bucket. Unfortunately it gives no valid reading while digging, since the shock loads causes the pendulum suspended in a viscous fluid to deviate from the correct position.

The potentiometer (connected to a little gearbox) was mounted on the drag drum. Being a voltage divider with the output voltage changing linearly when it is turned (while the input voltage is constant), it results in a reading that can be related to the number of turns of the drum from some reference position. The length of the drag rope can therefore be calculated as the drum is big enough so that the cable only forms one layer on the drum - the radius therefore never changes.

A custom built amplifier/power supply was purchased. It provides the excitation voltage for the loadcells, amplifying the resulting signal (full scale 5 Volts representing 500 kg) as well as being a power supply for the inclinometer and potentiometer and measuring the resulting signal. A digital readout with channel selector eased the calibration. The output from the amplifier/power supply was fed into a twelve bit PCMCIA card that plugged into a personal computer and the data was written to a file with a simple software programme. All readings were taken ten times a second and therefore a twenty hertz filter was build into the amplifier/power supply to avoid aliasing. This provided a continuous set of readings for all the measured values.

The loadcells were simply calibrated by zeroing and then hanging weights from it. This was repeated to assure repeatability, which was found to be within 1%. The potentiometer was calibrated by taking readings and measuring the distance from the fairleads to the drag hitch. A regression line was then fitted through these points, resulting in an equation relating a voltage to distance from the fairlead. To test for repeatability a piece of masking tape was attached to the drag rope just in front of the fairleads. The displayed measurement was then noted. The drag rope was paid out and pulled in till the same value was displayed on the digital readout. The position of the masking tape was noted and it was found that repeatability was very good (within about 10 mm). In order to check that the results for a specific test set-up would be reproducible, twenty runs were done, the results of which is summarised in Table 1. The values in brackets indicate what percentage or percentage change of the average a given value represents. It can be seen that the repeatability of results were good (especially the filling distance which is the most critical performance parameter) when considering the testing environment.

Table 1: Reproducibility of tests

	average	**standard deviation**	**minimum**	**maximum**
filling distance (m)	2.107	0.052 (2.47)	2.008 (4.7)	2.233 (6)
filling energy (J)	3780	149 (3.94)	3528 (6.7)	4048 (7.1)
drag force (kg)	384	16 (4.2)	357 (7)	410 (6.8)

5. EXPERIMENTAL PROCEDURE.

Swiericzuk (1994) who at that time was studying at the University of Queensland, where substantial research is being done on draglines states: "....Jeff Rowlands commenced work towards a PhD through The University of Queensland, studying Dragline Bucket Design (Rowlands, 1991). This was widely accepted as the first detailed report centred on the filling and design of dragline buckets." When searching for literature on draglines this statement was found to be valid. A lot of literature exists on draglines in general, but very little on the bucket itself. Another problem is that, since there are considerable profits to be made on dragline buckets, work that has been done on the subject is often confidential and therefore impossible to obtain.

It must also be mentioned that scale model testing of draglines is being done at a number of institutions. This proves that there is some merit in scale model testing. The University of Illinois has a training facility in which a 1:150 scale dragline is used and the image magnified onto a screen for training purposes (Anon., 1994). At the University of Queensland the large scale test rig and a model similar to the one at the University of Illinois exists (Sharrock et al., 1996). Scale model draglines (1:10 scale as well) are used by two Australian companies, ACIRL and Dragline Technologies, for research and training. It must however be stressed that these companies focus primarily on training and research, while Wright Equipment is believed to be the only bucket manufacturer with this ability.

5.1 Work done by Rowlands (1991)

In his report Rowlands (1991) on a number of occasions pointed out that there really has been done no comprehensive work on the dragline bucket itself, even though a lot of emphasis was placed on draglines in general. The need for further research on the subject was also emphasised. Because of the uniqueness of the report it was studied in great depth in order to set up a testing facility and an appropriate experimental procedure.

The work done by Rowlands (1991) included test work on an existing Small Scale Test Rig, which basically is a structure and motors simulating the drag and hoist functions of a dragline and allowed the simulation of the motion of the bucket during dragging and hoisting. The size of the bucket used was 1.3 litres. Thereafter a Two Dimensional Test Rig was built consisting of a bucket profile (slice taken out of the middle) running between two glass plates. This was done to observe material flow characteristics and helped in the development of the Shear Zone Theory.

The final stage of the project involved the construction of a Large Scale Test Rig which proved to be the most useful part for designing the experiments presented here since the raw data was included in the text. The aim his experiments was to determine factors that had a definite effect on the performance of a dragline bucket.

For these experiments two buckets were used - a Standard (STD) and a University of Queensland (UQ) bucket, with the UQ bucket having been wider and correspondingly shorter to obtain approximately similar volumes. They were tested in two different materials - gravel (Gravel) and decomposed granite (Deco). Short (S) and long (L) teeth were used in the experiments as well as a low (L) and high (H) angle of attack (only the short teeth were tested at a high angle of attack). Weights (a choice between one and two) were attached to the back of the bucket to correspondingly move the centre of gravity. The designation of the tests was done in the following consistent way - (bucket)(material)(length of teeth)(angle of attack of teeth)(number of ballast weights)(hitch height). A typical designation would be STD Deco SL12 - meaning the test was done with the standard bucket in the decomposed granite, using short teeth with a low angle of attack, with one weight attached to the back of the bucket and the drag chain in hitch number two.

As can be seen the factors that influence the performance of the bucket was investigated, but no effort was made to optimise any of these which was the main objective of this project when it was originated.

The work as presented in this thesis differs from that of Rowlands (1991) in the following aspects:

1. The geometry of the different buckets were comparable in that the relative position of the teeth, centre of gravity and hitch height were the same and the buckets were exactly the same weight and size.
2. Trends for changes in specific parameters were established more clearly. This was possible since more values of each variable were tested (for example four bucket widths instead of two) and these values covered a smaller range, causing the bucket to fill in all instances. In the case of Rowlands (1991) the spread was very, causing the bucket not to fill in a lot of instances and this prevented clear trends to be established.
3. Rowlands (1991) did not comment on the increase in required drag force that would be caused with his proposed changes. This is seen to be quite important and is discussed in detail in section 7.7.

5.2 Determining the parameters and sequence of testing

5.2.1. General discussion of factors involved

The digging cycle is a very complex and not well understood part of the excavation cycle. Then it also is true that there is a vast number of factors that influence this cycle and they can be split up into four categories - those that are site related, those that are machine related, those that are bucket related and those that are rigging related.

Factors that are site related:	drag angle, start drag distance from fairleads, type of material, size and distribution (fragmentation), angle of repose of the digging material, cohesion of the digging material, moisture content of the digging material.

Factors that are bucket related:	position of the drag hitch (vertically and horizontally), position of the hoist trunnion (vertically and horizontally), the position of the centre of gravity (vertically and horizontally), mass, length of bucket, width of bucket, height of bucket, length of teeth, angle of attack of teeth, number of teeth, etc.
Factors that are machine related:	available drag power drag speed and rated suspended load.
Factors that are rigging related:	the lengths of the drag and upper and lower hoist chain along with the dump rope length determine the carry angle, which was the most important rigging related parameter (see section 5.3).

The above is by no means a complete list of the factors influencing bucket performance. It just serves to illustrate that the number of factors that one ideally would like to test quickly becomes almost uncontrollable. Bearing in mind that wanting to do m variations of n variables requires n^m tests, it becomes clear that really only the most important factors should be included in the experiments.

5.2.2. Determining the importance of factors

According to Rowlands (1991) the single most important factor influencing the performance of the dragline bucket is hitch height. The tests performed by him (as discussed in section 3.1) were sorted into groups, for example STD Deco SL11 to STD Deco SL15 - a specific group therefore represents only a change in hitch height (from 1 to 5).

The raw data (Rowlands, 1991) was then used to determine the relative influence of the factors that had been tested. In each group the minimum filling distance and corresponding specific digging energy (SDE) were noted as well as the hitch height at which they occurred. The hitch height was therefore ignored when comparisons were made - thus in all comparisons an optimum hitch height was assumed (the optimum hitch height was noted however to allow for the observation of trends). This optimum could now be described by the following designation - STD Deco SL1. Note that the last number referring to the hitch height has been left out (the '1' therefore corresponds to one weight added to the rear of the bucket). The comparisons could then be made to these optimum values.

5.2.2.1 Influence of a change in bucket width

The UQ bucket was made 33% wider than the STD bucket. It is a big change, like all the others (as will be shown in sections 5.2.2.2), and it was concluded that comparing the different factors (even though not hundred percent correct) would give valuable information as to their relative influence.

For investigating the influence of change in bucket width, the data was compared as in Table A1, Appendix A. The bucket designation (UQ or STD) was taken out of the designation, leaving for example Deco SL1 (tests done in decomposed granite, with short teeth, low angle of attack and with one weight added to the rear of the bucket). The optimum values were then plotted in two columns, STD and UQ. The difference between the two buckets under the same circumstances (the only difference having been the hitch position which was taken as the optimum for each individual bucket) could then be compared.

The above was done for all the different groups. The difference in the filling distance and the SDE were then calculated for each group and the average was obtained. This average was used to determine the importance of the difference (in this case bucket width) - if it had a large value the difference was seen to be important and visa versa. The average difference for a change in bucket width was 0.52 meter for the filling distance and 19.87 for the SDE (specific digging energy (J/cubic meter)). The average difference in filling distance was about 20% which is substantial.

5.2.2.2 Rating of the importance of other factors that influence digging performance.

The other factors that were investigated were length of teeth, digging material, angle of attack and change in centre of gravity.

The teeth length was changed from 120 to 185 mm (54% increase). The difference in digging material could not be quantified but their physical properties were completely different. The one was gravel (like being used to built tar roads) distributed narrowly around a median particle of size of 14.3 mm (between 6 and 25 mm). The other was decomposed granite with a median particle size of 1.5 mm being distributed from fines (therefore high clay content) to almost 50 mm.

The angle of attack was changed from 45 to 55 degrees (22% increase) and the change in centre of gravity was obtained by adding an extra weight of 37.5 kg to the back of a 137.5 kg bucket (100 kg bucket plus one 37.5 kg weight that was always attached). The change in the centre of gravity therefore was also significant.

Similar tables (Table A2 to A6, Appendix A) as the one described in the previous section were drawn up for differences in teeth length, digging material, angle of attack and change in centre of gravity. The results (absolute values) are shown in Table 2:

Table 2: Rating of the importance of factors influencing bucket filling.

	Average difference in filling distance.	**Average difference in SDE**
Bucket width	**0.52**	**19.87**
Length of teeth	**0.34**	**18.15**
Digging material	**0.20**	**14.93**
Angle of attack	**0.16**	**7.03**
Change in COG	**0.02**	**8.01**

It can be seen that the bucket design (width) is by far the most important factor. The length of the teeth, the material in which the bucket digs and the angle of attack to a lesser extent also have a significant influence.

It can also be seen that a change in centre of gravity does not influence the performance of the bucket much - the optimum hitch height just changes to obtain a new equilibrium of forces. *Note: If the optimum hitch height is not changed the shift in COG would have a pronounced effect on the performance of the bucket.* It is also interesting that there is no evidence that the bucket filled better when it was heavier (two weights attached and hitch height changed to compensate for the shift in COG). This might lead to the conclusion that, in well-fragmented material, the bucket weight does not have a big influence on digging performance.

5.2.3 Sequence of testing

It was pointed out earlier that the number of tests needed to be done, to test a certain number of variables, increases dramatically as the variables increase. A fair number of variables have to be tested to obtain useful information however, and it was decided not to test all possible combinations of variables.

In the previous section the importance of the different factors influencing bucket performance was rated. The design of the bucket (width) seems to be the most important. It was therefore decided to build four buckets of different widths (see section 6.1) for details on the design of the buckets..

The length of the teeth was also seen to have a big influence, but since considerable effort and cost went into designing new teeth for the current dragline bucket, it was decided not to change the length of the teeth during the tests. Since the teeth are cast, changes are difficult and expensive and it was decided that changes in teeth length could be investigated at a later stage.

The influence of the soil type on bucket performance was also seen to be high. Obviously a bucket design that performs well over a range of conditions is desired. It was decided (see section 6.2) to test the bucket in two soil types. The first would be 26.5 mm crushed rock (no fines and very uniform) and the other 26.5 mm crusher run (top size 26.5 mm going down to fines). The crushed rock was chosen to simulate rocky conditions and the crusher run to establish the influence of fines. If deemed necessary a crusher sand could also be tested to monitor the performance in topsoil, but this accounts for only a small percentage of the operating time and was not included initially.

The angle of attack was also seen to have a fairly big influence and it was decided to test four different angles of attack. In addition the number of teeth could also be varied (5, 6 or 7 teeth). The position of the COG was seen to be not that important if the hitch was changed accordingly and it was decided to keep the position constant. The sequence for the tests, in each soil type, was planned as in Table 3:

Table 3: Sequence for tests

					FROM TESTS 1-20 GET WIDTH				TESTS 21-40 GIVE AA		
	W1	W2	W3	W4	AA1	AA2	AA3	AA4	NT1	NT2	NT3
HH1	1	6	11	16	21	26	31	36	41	42	43
HH2	2	7	12	17	22	27	32	37	44	45	46
HH3	3	8	13	18	23	28	33	38	47	48	49
HH4	4	9	14	19	24	29	34	39	50	51	52
HH5	5	10	15	20	25	30	35	40	53	54	55

In Table 3 HH1 to HH5 refers to the five hitches that was tested, W1 to W4 refers to the four bucket widths that was tested, AA1 to AA4 refers to the four angles of attack that was tested and NT1 to NT3 refers to the three different number of teeth that was tested.

Referring to Table 3 it can be seen that the four buckets (most important influencing factor) were tested for all five hitch heights. One bucket was then selected and the angle of attack was varied just for that bucket (testing all the hitch heights for every angle of attack). One angle of attack was then selected and the number of teeth changed for that angle of attack (testing all the hitch heights for every number of teeth). The above procedure would be followed in each soil type at a drag angle of 20 degrees (being a typical drag angle according to Rowlands (1991)). The data was evaluated continuously so that, if deemed necessary, extra tests could be performed. It was for example decided to test two of the buckets at a different drag angle, just to see what the influence on their performance would be, and observe differences in, for example, optimum hitch height.

The different values for each parameter are discussed in section 6.1 where the design of the test buckets is presented.

5.3 Testing procedure

In order to obtain meaningful results the testing had to be done very consistently. The testing sequence was explained in the previous section. The design of the test buckets and the soil selection were done very carefully to allow for meaningful and repeatable values and are explained in section 6.

The tests were done in the following consistent way. The optimum carry angle and the amount of material carried by each bucket were determined separately. The digging material was prepared to be exactly the same as for the previous test The bucket was positioned in the same starting position for each test. The drag ropes were then tightened just enough so that the bucket did not move. Dragging then commenced and continued until the bucket was full (a full bucket was defined as one where soil started to spill over the rear of the bucket). The data acquisitioning was done only for this part of the cycle. The bucket was then emptied and the bank was levelled to its original state.

The drag force on the SMD was not limited. When the bucket on a real dragline stalls the operator applies only a slight hoist force, with the result that the resistance force on the bucket decreases and it continues to fills. This procedure was avoided on the SMD since the human factor then was another variable that could not be monitored. The drag speed was also kept as constant as possible to 0.20 m/s (the max. drag speed on a Marion 8050 dragline is 2.5 m/s). The speed was calculated from the acquired data and in no test did the above average vary more than 0.05 m/s. Five tests were done for each different set-up to obtain a good average.

5.4 Additional tests

From the beginning it was realised that additional tests would probably be needed. Eventually the extra tests that were done were to establish the effect of a change in the drag angle and to establish the overburden carrying characteristics of the different buckets.

5.4.1. Change in drag angle

To establish whether a change in drag angle would have a significant influence on the optimum bucket configuration, it was decided to do some tests at a 30 degree drag angle instead of the normal 20 degrees suggested by Rowlands (1991). This was only done for two buckets in crushed stone and only with the angle of attack at 8 degrees and the number of teeth at 6. The results of these tests are discussed along with the rest of the test results in section 7.

5.4.2. Overburden carrying characteristics

Originally it was assumed that the optimum carry angle at pick-up would be equal to the drag angle, as suggested by Howarth et al. (1987). It was realised however that the different buckets might have different optimum carry angles, resulting in some of the buckets having been tested, and therefore having been set up, at an angle closer to their optimum. In order to gain some insight into the overburden carrying characteristics of the different buckets, the theoretical optimum for a simple two dimensional profile, having the correct height and length of the different buckets, was calculated. This was done for different angles of repose. Three graphs, each giving the theoretical optimum volume in the bucket as a function of carry angle for a specific angle of repose, are shown in Figure B1 and B2, Appendix B.

A comparison of the results is presented in Figure B3, Appendix B. From this figure the optimum carry angle appears to be fairly insensitive to a change in angle of repose even though the volume of overburden in the bucket is affected considerably. There also seems to be quite a difference in the amount of overburden carried by the different buckets, as well as a difference between the optimum carry angle of the buckets. Since the amount of overburden in the bucket is of great importance, it was necessary to do tests to see what the effect of the bucket dimensions is on the amount of overburden carried, as well as on the optimum carry angle.

The tests were only done in the crusher run (angle of repose between 38 and 42 degrees) and consisted of changing the dump rope length to obtain different carry angles at the pick-up point (with the drag angle staying constant). In section 3.5.1 it is discussed that the rigging design should allow the bucket to maintain it's carry angle at pick-up through to the dump point. The volume of overburden carried by the bucket at disengagement was then measured (after shaking the bucket in order to lose unstable material) as well as the carry angle of the bucket. Figures B4 to B7, Appendix B show the results obtained for the different buckets. It was difficult to obtain an accurate value for the overburden volume (it would probably be better to just weigh the mass of the overburden), but from the graphs it can be seen (ignoring the values for bucket 11) that there is not an appreciable difference in the amount of overburden carried by the different buckets. This contradicts the results of the two dimensional model as discussed above and suggests that a theoretical overburden carrying model will have to be more comprehensive. **The fact that the buckets were the same volume and carried the same amount of overburden resulted in the buckets eventually being compared on grounds of filling distance, filling energy and maximum required drag force alone.**

6. DESIGN OF TEST BUCKETS AND SOIL SELECTION

The design of the test buckets was done very carefully to allow for the comparison of results between the different geometries. The selection of the soil was also seen to very important in terms of modelling real operating conditions. The aim was to optimise the current dragline bucket. In scaling up the optimum design some caution must be applied however - it was stated earlier that the digging cycle is very complex and not well understood. When looking at it in terms of the possibility to destroy a reputation that took years to build, it is obvious that certain precautions will have to be taken in order to minimise risk involved in the scaling uncertainties. It was decided to design a 6 cubic metre dragline bucket according to the results from the tests performed and monitor the performance. By giving it for example three hitch heights the optimum hitch can be determined and compared to the results from the 60 litre bucket. Apart from minimising risk, additional insight would be obtained on the scaling effects. This was defined as a separate project and will not be documented here - it is mentioned only to show that, where knowledge might be lacking, an additional step will be taken to ensure good results.

6.1 Design of test buckets

When designing the test buckets, the reference was obviously the current design. From the start it was known that the current design performs very well in the field and big changes were not expected. The different values of the parameters were therefore centered around the current values. Figure 16 show a photo of the four test buckets and Figure 17 shows a side view comparison of the four buckets used in the project.

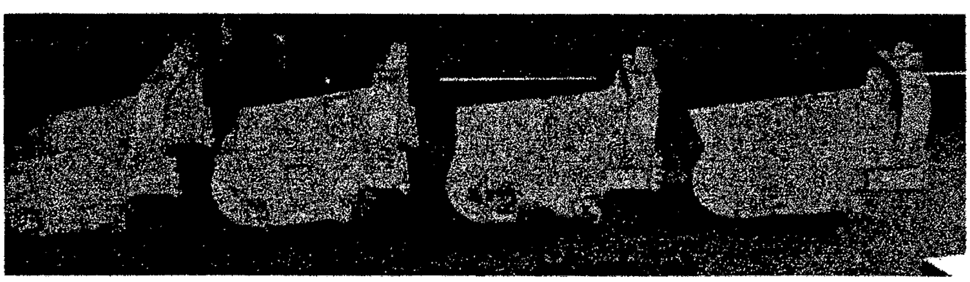

Figure 16: The four test buckets

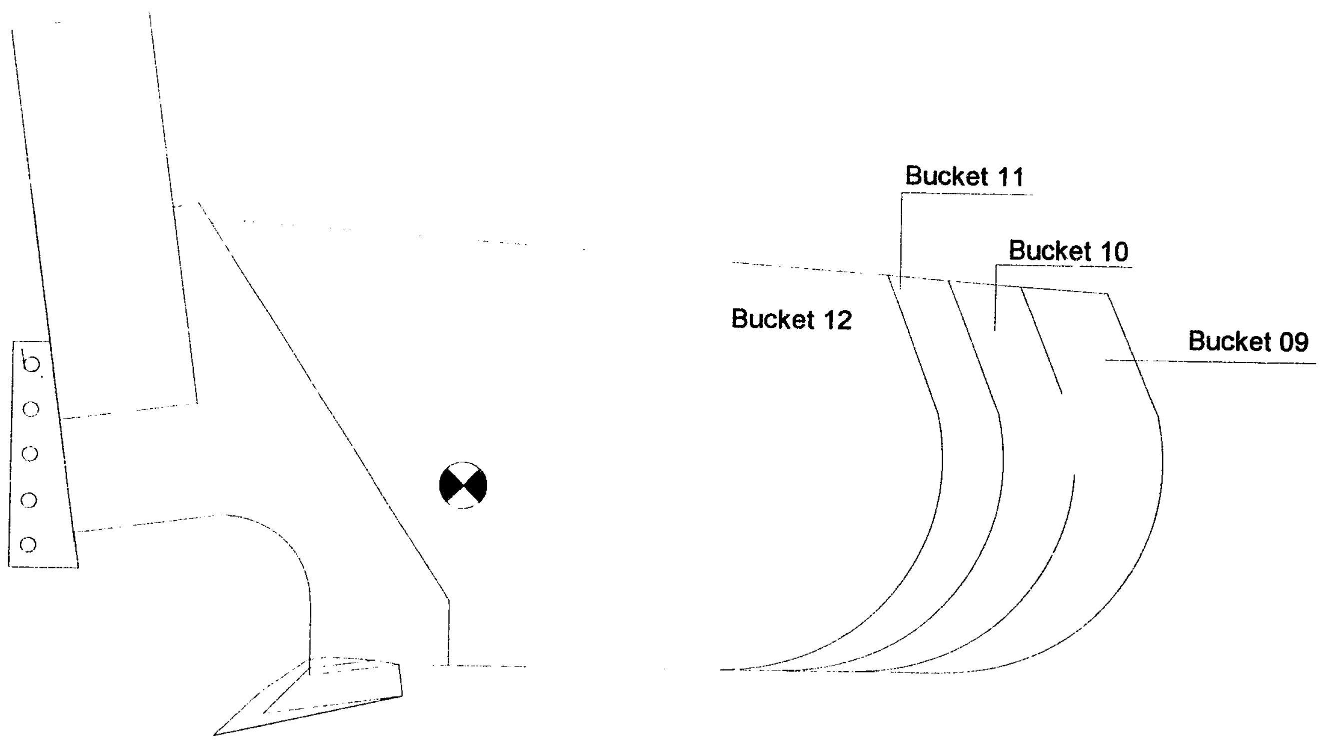

Figure 17: Side view comparison of the test buckets

The spread between the dimensions of the bucket was also not made to wide, since a test in which the bucket does not fill does not provide usefull information and the observation of trends cannot be done as effectively. This was realised when reworking the raw data from Rowlands (1991). In some instances only one of the five hitches he tested produced a full bucket and that made comparisons difficult. On the other hand the spread must be such that an optimum can be observed.

The buckets were designed to have the same struck volume. Rowlands (1991) used the following formula:

$$Volume = L_{ave} \times W_{ave} \times H_{ave} \times 0.9 \qquad (6.1.1.)$$

where

L_{ave} = average length

H_{ave} = average height

W_{ave} = average width

The calculated volume is the struck volume and the factor 0.9 is to account for the loss in capacity due to the curvature of the rear wall.

This approach was not considered accurate enough for this project (a difference in width must have an influence on the factor that account for the loss in capacity) and a detailed spreadsheet program was written to calculate the volume of the buckets exactly. This allowed the buckets to be designed to have exactly the same struck volume.

The front ring assembly (lip with teeth, cheekplates and arch) of the different buckets was designed to be the same apart from a difference in width. This was needed to be able to compare results obtained on the same hitch heights of different buckets. The wider buckets were then made shorter to have the same struck volume. The bucket widths that were chosen was the current width (scaled down 1/10), 0.9 times the current width, 1.1 times the current width and 1.2 times the current width. The designs were done on PRO ENGINEER, a solid modelling CAD package - the different buckets could then be viewed to ensure that the changes were significant but not unrealistic

On each bucket there were five hitch heights. They were the same on all the buckets due to the fact that the height of all the buckets was the same.

The mass of all four buckets was made exactly the same by adding weights. These weights were placed such that the centre of gravity of all the buckets was in the same place relative to the teeth and hitch position. The centre of gravity on the widest bucket was moved about 20 mm back and on the narrowest bucket about 20 mm to the front. It was shown earlier that a change in the position of the centre of gravity can be offset by the hitch position, without affecting the performance of the bucket much, and therefore the movement of the centre of gravity caused no concern.

The angle of attack (of the lip) on the current design is 8 degrees. The forward slope of the teeth is at an angle of 36 degrees to the top of the lip, causing an angle of attack of 44 degrees for the teeth. The angle of attack of the lip could be changed to 6,8,10 and 12 degrees respectively. This could be done easily by a bolt-on arrangement. The teeth were then connected to the lip in the usual fashion. The teeth were bolted to the lip and could be changed from the current number of 6 to either 5 or 7.

6.2 Soil selection

A lot of time and effort were spent on the selection of the soil. Ideally the soil should have the same physical properties (angle of repose, etc.) as the soil on the mine and should have the same influence on the scaled bucket as the real soil would have on the big bucket. That way scaling effects would be no problem. It must be remembered however that for each different soil type a truck load was needed and therefore laborotorium type activities would not be feasible.

To simulate the properties of the overburden accurately is an almost impossible task, since the number of variables is vast - size distribution, angle of repose, internal friction angle, cohesion, shear strength and moisture content, to name but a few. Obviously it is almost impossible to do tests for the whole series of variables. Furthermore it must be remembered that the ideal result of the whole exercise was to eventually incorporate all the knowledge obtained by the testing in a mathematical model (that did not form part of this project however). This means that the overburden characteristics must ideally be quantifiable.

A lot of research has been done on the cutting and excavation of rock. Nishimatsu (1972) represents a theory similar to cutting of metals for rock. A formula for cutting force is proposed and experimental work was done to establish the parameters in the equation. The cutting was done at a constant rake angle (angle of attack), which does not hold for dragline bucket operation.

Excavation and rippability indexes for intact rock have been established, but very little research has been done on blasted rock except stating that good fragmentation is very important. More of the research that was done on cutting of soil is discussed in section 8.

Considering all of the above, the following approaches where considered.

6.2.1 Obtaining soil from a mine

Going to a mine and grading a certain amount of overburden and scaling down the different size distributions does not seem to be a viable option. How much soil do you need to grade before the size distribution is representative? Remembering that the size distribution then is representative of the conditions of that specific spot of the specific mine - how many times will this exercise have to be repeated?

The approach followed by an Australian company ACIRL (who do similar tests with their scale model dragline) is to obtain soil from the mine and just remove the big rocks. This may sound like a good approach (or sales gimmick maybe?), but when the soil is blasted a certain amount of interlocking occurs which is destroyed when the soil is being worked for the first time (Basson and Chen, 1996). This seems to neutralise the effort to some extent. The soil then also is not as uniform as one might have hoped and this can result in tests not being reproducible. Results from tests performed by ACIRL were obtained and compared to results from the testing facility at Barlows (see Figure 18) - it is clear that the spread in results is much less in the case of Barlows and serves as proof for the selection of the soil that was finally made (see section 6.2.3).

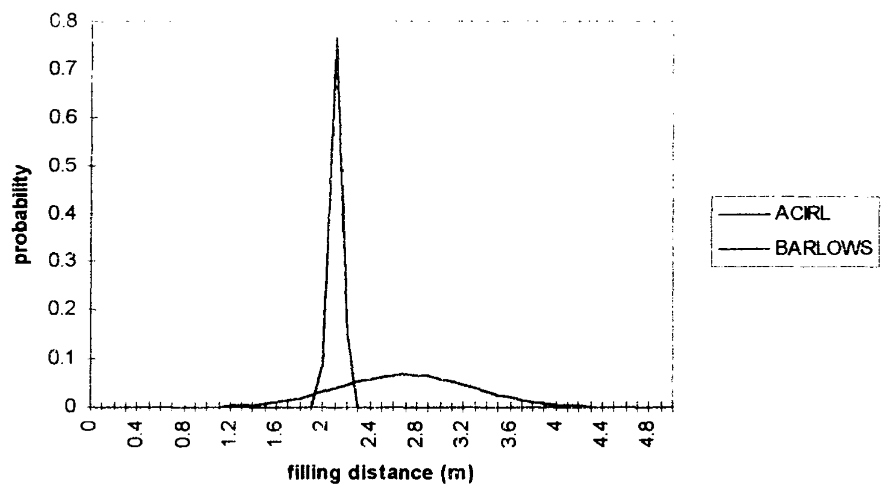

Figure 18: Comparison between tests performed by Barlows and ACIRL.

6.2.2. The Rosin Rammler distribution

Clearly a more manageable way of obtaining a fairly representative, repeatable and quantifiable overburden sample is required, remembering that an approximation really is the best that can be done.

The proposed model is one that is also being used by companies, like BLASTECH, who specialises in the blasting practices on mines. That is one of the main reasons for the selection of the model, as software is available with which the parameters for the model can be found from standard photographs of muckpiles at mines, enabling the quantification of the overburden at the mine. A so called Rosin Rammler distribution (Van Aswegen and Cunningham, 1986) can be used to represent the data. The formula that is used, is:

$$R = 100 \cdot \exp(-0.963 \cdot (\frac{x}{x_m})^n)$$

where

R is the percentage mass retained on screen size x.

n is the uniformity index: higher numbers represent more uniform muckpiles.

x_m is the mean size of the muckpile.

Thus the overburden type is characterised with two parameters. The shape of the particles, however, needs to be considered as well. Certain rock types have very characteristic shapes (irrespective of the size of the particles - it is a function of the chemical composition) and it was hoped that a fixed factor could be used to account for different rock types in the mathematical model. Heywood (1938) desribes a system that could be used. He presents the following formula:

$$k = \frac{k_e}{m\sqrt{n}} \qquad (6.2..2.1.)$$

where:

k: volume coefficient

k_e: volume coefficient for an equi-dimensional particle of the geometrical shape considered (0.5 average value)

$$m = \frac{width}{thickness}$$
$$n = \frac{lenght}{width}$$

With k_e varying very little and 0.5 a good average and determining m and n by measurement or visual inspection, the volume coefficient for each characteristic shape can be calculated.

A typical distribution was obtained from BLASTECH (mean size 35 mm and uniformity index 1.1). A soil composition was then made from available sizes from a crusher. The first reaction from most people who saw it, however, was that it was too rocky. Since there was not enough fines it also tended to "drain out" causing a variation in the conditions. According to Rowlands (1996) fines were very important and therefore this model failed. Cunningham (1996) also warned that application of the model is limited.

6.2.3. The selected option

Because of the failure of the previous models and in an attempt to find a viable solution, meetings were held with several geologists and mining engineers from the CSIR, none of whom thought there was a solution. At that point it was decided that an extra step would have to be included in the development process. That would consist of making design changes to a 6 cubic metre bucket. Monitoring the 6 cubic metre bucket would give a point in between the 60 litre and the 60 cubic metre bucket, to see what effect the scaling and the inability to scale the soil exactly, would have on the performance of the bucket.

It was then decided to compare the soil types only qualitati.ely (for example rocky, sandy, etc.) - one really wants a bucket that performs well over a range of soil types and therefore not scaling the soil exactly or not being able to quantify it might not be a big problem - as long as a bucket was found which performed well in a divergent range of soil types. The final selection that was made for the soil was the following:

1. **26.5 mm crushed rock:** It was to simulate rocky conditions
2. **26.5 mm crusher run:** It had a top size of 26.5 mm down to fines to establish the influence of fines.

The second selection was done due to a conversation with Rowlands (1996) in which he stressed that fines are important. If deemed necessary, a third soil type could be included - it would be crusher sand (just fines) and would help in determining the influence of rocks. It would also serve to simulate digging in top soil. Digging in top soil accounts for a small portion of the operating time and was therefore not included initially.

7. DISCUSSION OF RESULTS

As was discussed in the introduction and section 3.5.2.1, the aim of the project was to establish a geometry of the bucket that would allow it to fill in a shorter distance (in different materials) and with lower energy consumption than the current design. The maximum required drag force was not to increase as that could cause possible stalling of the bucket and longer fill times due to the characteristics of the drag motors. It must also be remembered that this was not exactly laboratory type experiments and that some variance was expected. Eventually the repeatability of the results turned out to be very good. About 120 sets of tests were performed, with each set consisting of 5 tests that were averaged, totalling to about 600 tests for which data were acquired and evaluated.

When the buckets were designed, it was kept in mind that really only tests that produce full buckets were of interest, since that would allow comparison of different geometries while a geometry that did not produce a full bucket was of no use as far as comparisons were concerned. Eventually the bucket did fill in all cases (although taking long to do so in some cases) which was considered to be a good result in itself.

It must also be admitted that the tests as were done here are very similar to those done by Rowlands (1991). He however attempted to show which parameters influence dragline bucket design, with no effort to optimise them. In this project four buckets instead of Rowlands' two were used and the relative position of the teeth, hitch and centre of gravity on all the buckets were the same, which allowed direct comparison of the same geometry on different buckets. Rowlands (1991) noted, and it is widely accepted, that the hitch height is the single most important design parameter on the dragline bucket. Therefore all comparisons that follow are made over the range of hitch heights. Rowlands (1991) also encountered a number of cases in which the bucket would not fill and that made any comparisons difficult. He mentioned that the wider bucket outperformed the narrower one, but he did not comment on the increase in maximum required drag force and how to possibly counter that. The differences between the tests documented here and those by Rowlands (1991) is presented in section 5.1.

All figures related to this section are included in Appendix C, since the number makes it impractical to present them here. The drag force is given in kilogram instead of newton, since in Wright Equipment there is a preference to express drag force in tons. A kilogram value on a tenth scale model, scaled according to equation 4.2.2.2, then have the same numeric value as the ton value on the prototype (for example 280 kg and 280 tons respectively).

7.1. Comparison of different bucket widths

The tests as discussed here were done for all four buckets in both digging materials, using all five hitches. In all cases 6 teeth and an angle of attack of 8 degrees were used. The filling distance will be discussed first in all cases, as that is the most important performance parameter.

Filling distance: Figures C2 and C3 show the filling distance as a function of hitch height for the different buckets in crushed rock and crusher run, respectively. It can be seen that all the buckets filled the quickest in hitch height three (in both materials). With the buckets having rather different geometries, but with the centre of gravity in the same position relative to the hitch and teeth, it suggests that good filling is a function of the position of the teeth, hitch and centre of gravity. It also suggests that good filling is a function of how the bucket engages initially. This was expected, since, when reworking the data of Rowlands (1991), it was noted that the shortest filling distance was almost exclusively obtained in the hitch where the engagement rate (rate of increase of force as function of distance at engagement) were a maximum for a specific geometry (see Rowlands, 1991, Table A7, Appendix A). With the engagement rate being a function of how the bucket engages, it was realised that initial engagement of the bucket played a major role in obtaining good filling distances.

It is also noticeable in both digging materials that the wider the bucket, the quicker it filled (bucket 12 being the widest and bucket 09 the narrowest bucket). Making the bucket wider will have some structural implications as discussed in section 7.8.

Maximum drag force: Figure C1 shows a typical graph of the difference in filling in different materials. The two runs are for exactly the same configuration in crushed stone and crusher run respectively. It can be seen that the filling distance in the crusher run (fines) is less, but the maximum drag force is higher. Considering the area under the graph it also seems as if the filling energy is more in the case of the crusher run (fines). Noticeable as well, is the fact that the amount of shock loading is much less in the case of the crusher run (even though the force is higher) as was observed with measurements on real dragline buckets. This is fortunate since the amount of shock loading in the crusher run made it difficult to obtain good values for the maximum required drag force. With the crusher run resulting in less shock loading, good values for the maximum required drag force could be obtained and these were the important values, when considering stalling of the bucket, since they were the highest (see section 7.2).

Considering Figures C4 and C5 which show the maximum required drag force as a function of hitch height for the different buckets in crushed stone and crusher run respectively, it can be seen that no trend can be picked up in the crushed stone as far as bucket width is concerned. In the crusher run (were the good values could be obtained due to less shock loading) it can be seen that the wider the bucket however, the higher the required drag force is. This relation is quantified in section 7.7. From both graphs it can be seen that the maximum required drag force increases rapidly as the hitch height increases. This relation is also quantified in section 7.7 and recommendations are made in section 7.8.

Filling energy: Considering Figures C6 and C7 which show the filling energy as a function of hitch height for the different buckets in crushed stone and crusher run respectively. In both cases it can be seen that the filling energy is fairly constant over the lower three hitches (filling distance decrease and drag force increase), and increases over hitches three to five (filling distance and drag force increases. It can also be seen that the wider the bucket, the less energy is required to fill it.

From the above it can be seen that a wider bucket outperforms a narrower bucket as far as filling distance and energy is concerned, but that it also requires a higher drag force to fill it, which might cause the bucket to stall. The stresses on the drag equipment will also increase, which might lead to increases in maintenance and downtime. A compromise for this situation is discussed in section 7.8.

7.2. Comparison of different digging materials

The tests as discussed here have been done for all four buckets in both digging materials, using all five hitches. In all cases 6 teeth and an angle of attack of 8 degrees were used. It is the same results that have been discussed in the previous section, but is presented differently.

Filling distance: Figures C8 to C11 show the filling distance as a function of hitch height for both digging materials for buckets 12, 11, 10 and 09 respectively. Apart from hitches one and two on bucket 09 and hitch five on bucket 12, it can be seen that the filling distance in the crusher run (fines) is consistently shorter than for the crushed stone. The optimum hitch in all instances is however the same, which is a good result and indicates that a bucket geometry can be selected that will work well over a range of digging conditions.

Maximum drag force: Figures C12 to C15 show the maximum required drag force as a function of hitch height for both digging materials for buckets 12, 11, 10 and 09 respectively. It can be seen that the force is consistently higher in the case of the crusher run (fines) for all buckets and on all hitches. This validates the use of the values as obtained in the crusher run for purposes of stalling.

Filling energy: Figures C16 to C19 show the filling energy as a function of hitch height for both digging materials for buckets 12, 11, 10 and 09 respectively. It can be seen that the values in the crusher run (fines) are consistently higher than the values in the crushed stone for all the buckets and on all the hitches.

From the above it can be seen that, while overburden that contain a high percentage of fines is normally considered easy digging (probably due to the shorter filling distance), the maximum required drag force is actually higher than required in rocky conditions. If the rocks get big however, it will probably be found that the required drag force increases, even though the amount of fines is very little. The high amount of shock loading encountered in rocky conditions should be taken into account when designing against fatigue. It can also be seen that a bucket geometry that works well in a range of overburden types can be selected, as long as the fragmentation on blasting is good.

7.3. Comparison of different angles of attack

These tests were done for only two cases. Firstly it was done for bucket 10 in crushed stone, using angles of attack of 6, 8, 10 and 12 degrees. The results from this test led to the fact that fewer tests were done in the second case. The tests in the second case were done with bucket 11 and using only angles of attack of 8 (standard) and 12 in the crusher run to see whether the same observations could be made for quite different conditions (different bucket and different digging conditions). In all cases 6 teeth were used.

Filling distance: Figures C20 and C21 show the filling distance as a function of hitch height for different angles of attack for bucket 10 in crushed stone and bucket 11 in crusher run respectively. From figure C20 it can be seen that lowering the angle of attack from 8 to 6 degrees caused the optimum hitch height to shift from position 3 to 4. Conversely a change in the angle of attack from 8 to 12 degrees caused the optimum hitch height to shift from position 3 to 2. It is also very important to note that the optimum filling distance for all angles of attack is about the same (even though obtained in different hitch positions). This led to the conclusion that an increase in the angle of attack causes the optimum hitch height to shift lower and a decrease causes the optimum hitch to shift higher, but no gains are to be made on filling distance. Therefore the only motivation for increasing the angle of attack would be to move the hitch lower where the maximum required drag force is less. Once this was established it was decided to do the tests with bucket 11, in crusher run, only for angles of attack of 8 and 12 degrees to see whether the same observation could be made.

Figure C21 shows the filling distance for bucket 11 in crusher run as a function of hitch height for angles of attack of 8 and 12 degrees. It can be seen once again that the increase in the angle of attack caused the optimum hitch to shift down one position without changing the filling distance. It can therefore be concluded that the optimum hitch height can be shifted down by increasing the angle of attack. The effect on the drag force will be discussed next, because a lower hitch is associated with a lower required drag force, but the change in the angle of attack might also have an effect in that one feels it will increase the drag force and thereby offset the advantage that was seeked.

Maximum drag force: Figures C22 and C23 show the maximum required drag force as a function of hitch height for different angles of attack for bucket 10 in crushed stone and bucket 11 in crusher run respectively. It can be seen that the increase in angle of attack causes the required drag force to increase slightly for the same hitch height, but the combination of a higher angle of attack and lower hitch height (AA12 and HH2 vs AA8 and HH3) leaves the filling distance unchanged, lowers the drag force by 13.8% and decreases the filling energy by 4.2%.

Filling energy: Figures C24 and C25 show the filling energy as a function of hitch height for different angles of attack for bucket 10 in crushed stone and bucket 11 in crusher run respectively. It can be seen that the difference in filling energy between the different angles of attack is small, especially in the lower hitches which is of interest (generally the filling energy and the maximum required drag force is less).

The above goes to show that changing the angle of attack will not have a big influence on the productivity of the bucket as a whole, ***provided that the hitch is changed accordingly.*** Changing only the angle of attack on a bucket will have a big influence on the productivity of the bucket. Considering the increased loading on the lip of the bucket, which is the most highly stressed area on the bucket as it is, the decrease in drag force obtained by a higher angle of attack and lower hitch does not seem worthwhile. It must be remembered that Rowlands (1991) changed the angle of attack by 10 degrees and even then it was shown in section 5.2.2.2 (Rowlands, 1991) that the hitch could to some extent offset the change, which serves to validate the test results.

7.4. Comparison of different numbers of teeth

The different numbers of teeth that were tested, were tested on bucket 10 for two angles of attack (8 and 12 degrees) and only in the crushed stone to see whether any trends could be observed.

Filling distance: Figures C26 and C27 show the filling distance as a function of hitch height for different numbers of teeth on bucket 10 in crushed stone for angles of attack of 8 and 12 degrees respectively. It can be seen (especially for the 8 degree angle of attack) that the differences were small. The optimum hitch height for example did not change and neither did the filling distance in the case of the 8 degree angle of attack. Bigger variances on the higher angle of attack can be expected since the projected vertical area of each tooth is more and an added tooth results in a bigger change.

Maximum drag force: Figures C28 and C29 show the maximum required drag force as a function of hitch height for different numbers of teeth on bucket 10 in crushed stone for angles of attack of 8 and 12 degrees respectively. It can be seen that the differences are small. In the case of the 12 degree angle of attack (where adding a tooth does have a bigger difference than on the 8 degree angle of attack) it can be seen that the added tooth will cause only a slight increase in the required drag force as would be expected.

Filling energy: Figures C30 and C31 show the filling energy as a function of hitch height for different numbers of teeth on bucket 10 in crushed stone for angles of attack of 8 and 12 degrees respectively. It can be seen that the differences were small.

From the above it can be seen that the influence of the number of teeth is small. It seems as if an extra tooth might cause a slight increase in the required drag force, but on the standard 8 degree angle of attack this seems to be negligible small.

7.5. Comparison of different drag angles

At this stage of the project a fair idea of the influence of the different factors has been obtained, but it was realised that some extra tests that were not originally included in the testing sequence, would have to be performed. It was considered essential to see what influence a change in the mine layout would have on a bucket. It was thought to establish this by changing the drag angle (which is more or less equal to the angle of the slope). Buckets 10 and 12 were then tested at a new drag angle of about 30 degrees (the other tests having been done at a drag angle of about 20 degrees) in crushed stone only.

Filling distance: Figures C32 and C33 show the filling distance as a function of hitch height for two different drag angles on buckets 10 and 12, respectively, in crushed stone. It can be seen that the filling distance for both buckets on all hitch heights are less for the higher drag angle. This was expected since the added gravity will aid the flow of material into the bucket. It is however interesting to note that the optimum hitch height does not change for either of the buckets. This result can not be related to digging on uneven terrain (which would be almost impossible to model consistently), but leads to the conclusion that, while digging on straight slopes, a bucket that performs well at one drag angle will generally perform well at all drag angles. It can also be seen that productivity will be higher while digging steep slopes. It also seems as if the difference in the filling distances of the two buckets is less at the higher drag angle. This could mean that the advantages associated with a wider bucket is less at higher drag angles.

Maximum drag force: Figures C34 and C35 show the maximum required drag force as a function of hitch height for two different drag angles on buckets 10 and 12, respectively, in crushed stone. The conclusions to be drawn from the different buckets are contradictory and no clear trends could be ascertained. The diffrences in drag force were small and might suggest that the drag force is influenced weakly by drag angle.

Filling energy: Figures C36 and C37 show the filling energy as a function of hitch height for two different drag angles on buckets 10 and 12, respectively, in crushed stone. It can be seen that for both buckets, on all five hitches, the filling energy was less for the higher drag angle. This was expected, since filling at higher drag angles requires similar drag forces, but shorter filling distances.

All of the above show that, while the bucket fills easier (shorter filling distance and lower filling energy) at the higher drag angles, the optimum bucket width does not change, but the advantage that the wider bucket has over the narrower bucket is less, since the difference in filling distance is less and the filling cycle is a smaller percentage of the cycle as a whole. The optimum hitch height doesn't change either, which leads to the conclusion that one geometry could perform well at different drag angles.

7.6. The effect of changing the hitch position horizontally

These tests were also not originally included in the testing sequence Since it was known from the start that the hitch height was the single most important design parameter on the dragline bucket, it was decided to see what the difference would be of shifting the hitch forward and backward. This was done to establish whether only hitch height or hitch position in general was the important design parameter. These tests were only done for bucket 11, with an angle of attack of 8 degrees and 6 teeth, in crusher run (fines). The hitch was moved 30 mm forward and 30 mm back. In the backward position only the bottom three hitches could be drilled because of interference from the arch, but, since it was established earlier that the lower hitches are really the ones of interest and the optimum was expected around hitch 3, this was thought to be good enough.

Filling distance: Figure C38 shows the effect on the filling distance when the hitch position is shifted forwards and backwards. The tests were done for bucket 11 in crusher run (fine with an angle of attack of 8 degrees and 6 teeth Especially when considering the 30 mm forward and the 30 mm backward positions (these were done weeks apart from the standard) and when considering the values at the optimum hitch (hitch 3) it can be seen that the effect of shifting the hitch forward and backwards is very little. It is also noticeable that the optimum hitch height for all three cases is hitch height 3, which indicate that a horizontal shift in the hitch position does not have a big effect

Maximum drag force: Figure C39 shows the effect on the maximum required drag force when the hitch position is shifted forwards and backwards. The tests were done for bucket 11 in crusher run (fines), with an angle of attack of 8 degrees and 6 teeth. It can be seen that the differences between the forward and backward hitches are very small.

Filling energy: Figure C40 shows the effect on the filling energy when the hitch position is shifted forwards and backwards. The tests were done for bucket 11 in crusher run (fines), with an angle of attack of 8 degrees and 6 teeth. Once again it can be seen that the difference between the forward and backward hitch position is small.

From the above it can be seen that shifting the hitch horizontally does not have a big influence on the performance of the bucket.

7.7. Quantifying the effect of bucket width and hitch height on the maximum required drag force

It has been seen that, in order to improve performance, the bucket should be made wider. This could cause the bucket to stall however, since the required drag force increases with bucket width. It has also been seen that a way of decreasing the required drag force is to lower the hitch. Further, it was shown that changes in the angle of attack, number of teeth and shifting of the hitch in a horizontal direction does not have a big influence on bucket performance.

The effect of bucket width was then quantified along with the effect of bucket width on filling distance and filling energy. Figures C41 and C42 show this effect for crushed stone and crusher run respectively for the different buckets, using hitch height 3. Figures C43 and C44 show this effect for crushed stone and crusher run respectively for the different buckets, using hitch height 2. Hitches 2 and 3 were used because this is the hitches that are considered important - short filling distance, low filling energy and fairly low maximum required drag force. In all cases the correlation coefficient is high (above 0.85), meaning that the linear approximation is a good assumption. Note that the relationship for drag force was only quantified in the crusher run, since it is the material in which the drag force values were the highest, and therefore critical, and the case in which good values could be obtained. In all cases it can be seen that widening the bucket causes the filling distance and energy to decrease. Considering Figures C42 and C44, it can be seen that for both hitch 2 and 3 an increase in bucket width of 20% will have the effect of increasing the maximum required drag force by 100 kg (the slope of 500 times 0.2). Figures C45 and C46 show the effect of hitch height on filling distance and drag force respectively for all the buckets in crusher run **(considering only the lower three hitches as those were considered important - these approximate linear relationships therefore only hold over the three lower hitches)**. It can be seen that for all the buckets the filling distance increases as the hitch is changed from 3 to 1, but it can also be seen that bucket 12 in hitch 2 filled quicker than bucket 10 in hitch 3, even though hitch 2 is not the optimum hitch position for bucket 12. From Figure C46 it can also be seen that, considering the slope of the regression lines for bucket 10, 11 and 12 (which are all fairly close to 90), a decrease of one in hitch height will cause the required drag force to decrease by about 90 kg. Therefore it can be seen that increasing the bucket width by 20% will have about the opposite effect on the required drag force as lowering the hitch by 30 mm (300 mm on a real bucket).

It is therefore clear that by widening the bucket and lowering the hitch, the required drag force can be kept constant and, as was seen from Figure C45, the filling distance can be decreased. To prove this point the values as obtained for bucket 10 in hitch 3 and bucket 12 in hitch 2 (which means the hitch was made 30 mm lower (300 mm on the real bucket) and the bucket width was increased by 20%) was compared in Table 4 (Note that the drag force values for the crushed stone were not included due to the amount of shock loading and since it was lower than the values in the crusher run, as was discussed in section 7.1):

Table 4: Comparing the values obtained with bucket 12 in hitch 2 with the values obtained with bucket 10 in hitch 3 as the standard.

	crusher run (fines)	crushed stone
decrease in filling distance	16%	9%
decrease in filling energy	20%	20%
decrease in max. drag force	-1.7%	

The values as obtained for bucket 10 in hitch 2 and bucket 12 in hitch 1 (which means the hitch was made 30 mm lower (300 mm on the real bucket) and the bucket width was increased by 20%) are compared in Table 5:

Table 5: Comparing the values obtained with bucket 12 in hitch 1 with the values obtained with bucket 10 in hitch 2 as the standard.

	crusher run (fines)	crushed stone
decrease in filling distance	13%	9%
decrease in filling energy	21%	20%
decrease in max. drag force	7%	

Both of the above cases represent an increase of 20% in bucket width and a lowering of hitch height of 300 mm, from two different positions, and the results are given in two digging materials. In both cases it can be seen that performance of the bucket is improved considerably.

7.8. Conclusions regarding the influence of geometry on dragline buckets

In section 7.1 to 7.7 it was shown that no real benefit could be obtained by changing the angle of attack or the number of teeth. It was also shown that making the bucket wider and shorter (obtaining the same struck volume) and lowering the hitch (to ensure that the required drag force does not increase) results in improved performance. When comparing the **WRIGHT BUCKET** to other buckets on the market it can be seen that it is generally a shorter and wider bucket. It is believed that this, along with the fact that it is a lightweight bucket (therefore bigger capacity for the same suspended load) is the reason for the excellent performance of the bucket on the mines. The increased performance has been established by feedback from the mines and was confirmed by tests that was done by Lumley and Jensen (1996).

There are however problems associated with the proposed change. A wider bucket will result in higher stresses on the lip, which is highly stressed as it is. The arch will also be higher stressed - an area in which structural problems were experienced originally. The cross sectional area of the lip and arch will therefore have to be increased, resulting in added weight, which is what should be minimised on a lightweight bucket. Higher dump rope loads is also expected with shorter buckets because of the shorter moment arms (Lumley and Jensen, 1996) and the dump rope loads of the **WRIGHT BUCKET** is higher than most other buckets as it is (Lumley and Jensen, 1996). The stability of wider buckets is also less while swinging and nodding and overshooting at disengagement is more pronounced. Shortening the bucket will also cause the load carrying characteristics (from disengage to dump point) to change (which is very good for the current Wright bucket, according to Lumley and Jensen, 1991) as well as influence the dumping characteristics. When the hoist trunnions are positioned to far back, the bucket will dump prematurely and when they are positioned to high, the bucket will not dump cleanly - this was found to happen when trials were done on BE buckets (Ferreira, 1997).

With a lower hitch, one would also expect more wear on the drag chains, which will result in increased maintenance time and costs. On mines employing the extended bench mining method, the extended bench is usually build from the material from the keycut, which is the material that is rehandled. The keycut is one bucket width and an increase in bucket width could result in increased rehandle on these mines, which might offset the advantages to be gained from a wider bucket.

The advantages and disadvantages of a wider bucket can be summarised as follows:

- **Advantages**
    - Shorter filling distance (10 to 15%)
    - Lower energy consumption (20%)

- No increase in required drag force
- **Disadvantages**
    - Higher stresses
    - Higher dump rope loads
    - Less stability while swinging
    - Nodding and overshooting
    - More wear on drag chains

It has been proven by the **WRIGHT BUCKET** that an increase in productivity is possible by widening the bucket and it must be remembered that small increases in productivity result in huge financial gains. In section 9 (Conclusions and Recommendations) research needed to address the problems associated with wider buckets is discussed.

8. UNDERSTANDING THE DIGGING PROCESS

This chapter is included to try and explain some of the trends that was observed The literature concerned with digging or cutting of rock is presented here to show that, while rock cutting (similar models to metal cutting at constant rake angle) and excavation of unblasted rock have received attention, no work has been done on understanding the digging cycle of a dragline.

The digging process is very complicated, since the bucket has got 6 degrees of freedom (DOF) and that in a non-uniform material, while the starting conditions are variable. From a numeric modelling point of view this seems like a mammoth task to model. In fact most kinds of soil engagement machinery (of which the dragline is probably the most complex) have defied analysis (Reece, 1984). He also mentions that to progress one should continuously try to understand, in quantitative ways, the action of machine elements, which is part of what was attempted in this project.

Some research has been done on the cutting of rock and soil engagement machines. Nishimatsu (1972) did tests and developed a theory for the cutting of rock that is very similar to cutting of metal during machining operations. The cutting tool had only 1 DOF, the rake angle was constant and the material uniform. Spektor and Katz (1985) did experimental work on frontal resistance force in the cutting of soil. It consisted of knives having 1 DOF that were forced through soil. They found that the frontal resistance force was linearly related to tool width as was found in this project. A number of papers attempting to assess ease of excavation of unblasted rock (based on properties like Uniaxial Compressive Strength, Schmidt Hardness Value, Seismic Velocity and taking into account factors like weathering) were found (Hadjigeorgiou and Scoble, 1990; Karpuz, 1990 and Panagiotou, 1990). Bolukbasi et al., (1991) describes the O&K wedge test that is used to determine the mechanical cutting characteristics of rocks for selection of bucket wheel excavators.

Specific digging energy is commonly accepted as a measure of cutting efficiency or diggability of soil ((Panagiotou, 1990), (Bolukbasi et al., 1991) and (Ceylanoglu et al, 1994)).

Fragmentation is of great importance for dragline performance. Michaud and Blanchet (1996) found that better fragmentation lead to better mine productivity, but blasting and drilling costs increased. The higher productivity is because of better loading and because of higher densities associated with good fragmentation.

Howarth et al. (1987) did dragline bucket filling tests on a simulator having 1 DOF at the University of Queensland. The work of Rowlands (1991) followed on that project and he mentions that the set-up as was used by Howarth et al. (1987) was not representative of real dragline operation.

8.1. Theoretical static analysis of the forces on the bucket at engagement

This section is aimed at understanding the digging performance qualitatively. As was discussed earlier, digging performance is a function of initial engagement and more specifically of the relative position of the hitch, teeth and centre of gravity. It was believed that considering the forces on the bucket at engagement could provide some insight into bucket performance. Figure 19 shows the bucket at engagement as well as the forces on it. The magnitude and direction of the forces on the teeth will be a function of the position of the COG, the hitch position and the tilt angle. The force diagram on the bucket was then solved statically for different hitch positions and tilt angles. The bucket was assumed to rotate around the tip of its cutting edge after the teeth engaged. Figure 20 shows the magnitude of the force on the teeth and its direction (angle between centreline of teeth and force), for different hitch positions and tilt angles. **This is the force neccessary to keep the bucket in equilibrium.**

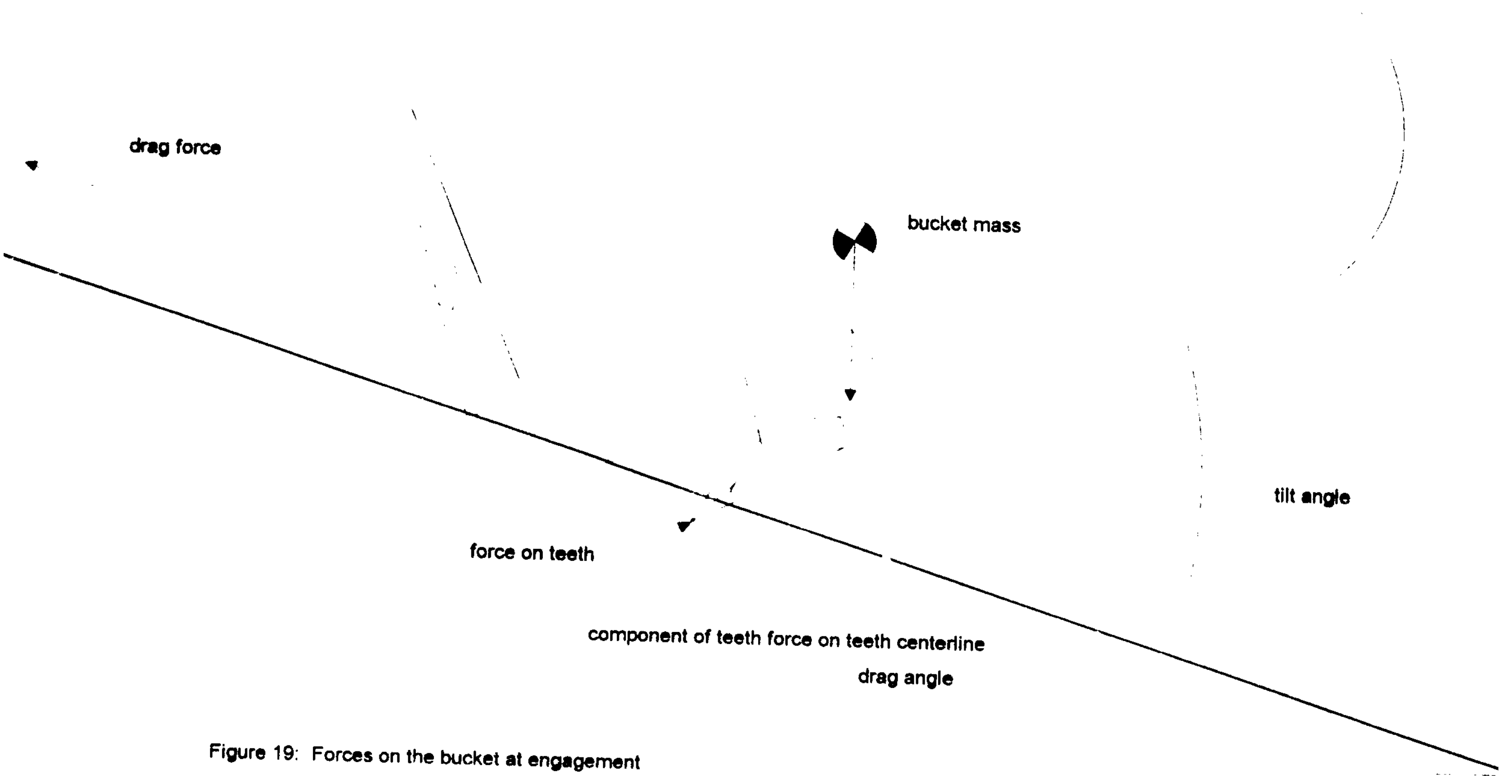

Figure 19: Forces on the bucket at engagement

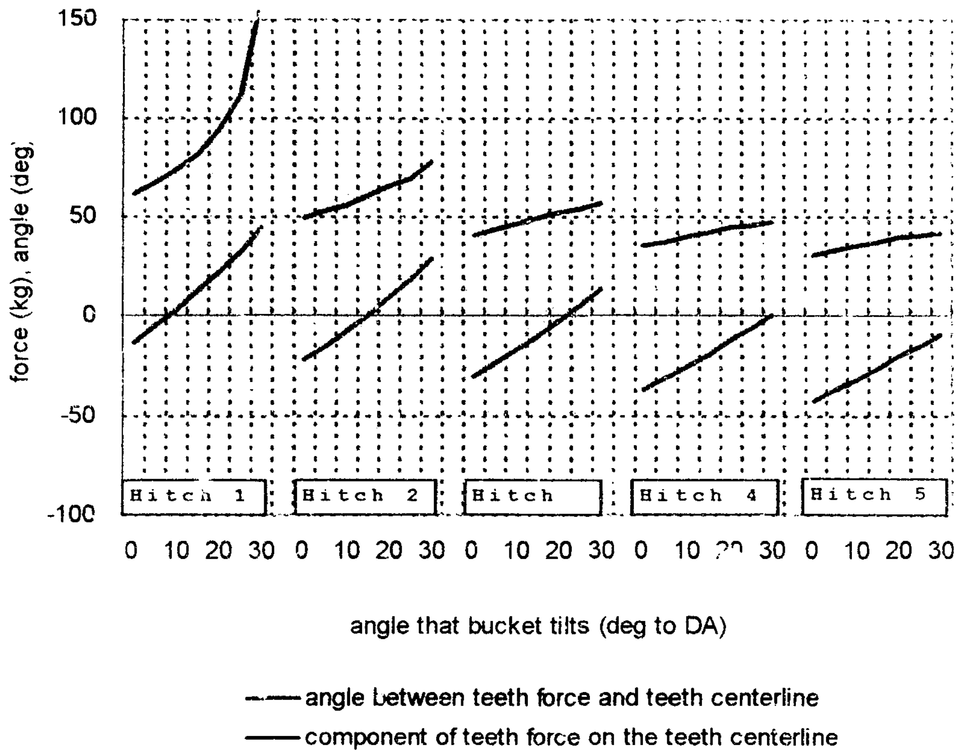

Figure 20. Theoretical teeth force at engagement

8.2. Qualitative assessment of the influence in change of parameters on digging performance

From Figure 20 a number of interesting observations can be made, even though it is just an approximation. It can be seen that the force on the teeth at low hitch heights is high, even at low tilt angles. This means that the force the digging material needs to exert on the teeth to keep the bucket in equilibrium (to keep the back up) at that tilt angle is high. If that force can not be exerted on the teeth, the teeth will shear the soil and the tilt angle will remain very low. This will result in the bucket skidding over the material as was observed at low hitch heights.

From the figure it can also be seen that the force on the teeth increases as the tilt angle increases. It is believed that the bucket tilts forward until the force on the soil is enough for the teeth to shear the soil. Assuming that the force to shear rocky material is more than to shear sandy material (it is easier to push a shovel into sand than into a pile of stones), this could be the reason why a bucket would tend to have higher maximum tilt angles in rocky material than in sandy material. It can also be seen that the force on the teeth at high hitch heights is low for all tilt angles. Therefore, no matter how far forward the bucket tilts, the force on the soil is never enough to shear it, resulting in the bucket falling forward.

It is also noteworthy that in the optimum hitch (hitch 3) the angle between the force on the teeth and the teeth centreline is less than 20 degrees for all tilt angles. This means that 0.94 (cos 20) times the magnitude of the force is always available as a penetrating force and it can be seen that this does not hold for the other hitch heights.

In the tests it was found that an increase in the angle of attack, resulted in the optimum hitch being lower. It could be explained as follows. Increasing the angle of attack results in the resistance force on the teeth at zero tilt angle (from where the bucket starts) to be higher, since the frontal area is more. The material will not shear as easily and the bucket will tilt forward and engage positively even when a higher force is required on the teeth to keep the bucket in equilibrium. It can be seen from Figure 19 that a lower hitch height requires a higher force on the teeth to keep the back of the bucket up and it can be concluded that the optimum hitch height is lower because the resistance force that can be exerted on the teeth is higher for the higher angle of attack.

9. CONCLUSIONS AND RECOMMENDATIONS

This thesis documents the experimental work that was done in developing a geometry for a dragline bucket that would fill quicker and fill with lower energy consumption while the maximum drag force does not increase. For the purposes of tests, a scale model dragline was build, which accounts for a lot of effort going into the project as a whole. Four test buckets of different geometries were tested. The work that was done can be seen to be an extension of the work done by Rowlands (1991). More buckets were tested, however, and the design of the buckets (position of centre of gravity, etc.) was done more carefully. This allowed for the observation of trends. In addition, the relationship between angle of attack and hitch position was established, and the hitch position to offset the increased drag force associated with a wider bucket was found. The relationships between maximum drag force, filling distance and filling energy as a function of bucket width and hitch height were quantified in crusher run.

It was found that the most significant increase in productivity could be obtained by making the bucket wider. This increased the required drag force. The required drag force was found to increase rapidly with an increase in hitch height and lowering the hitch, therefore, could offset the increase in drag force. The centre of gravity of all the buckets were in the same place relative to the hitch and teeth. With all the buckets filling the quickest in the same hitch, it can be concluded that the relative position of the teeth, centre of gravity and hitch determines the filling characteristics to a high extend and also that initial engagement is the key to good filling performance. Lowering the hitch to offset the increase in drag force associated with a wider bucket, would therefore result in the bucket not operating in its optimum hitch, but significant increases in performance could still be established - about a 15% reduction in filling distance and a 20% reduction in filling energy.

These changes does have problems associated with them. A wider bucket will result in higher stresses in the lip (which is highly stressed on any bucket) and a lowering in hitch height will result in more wear on the drag chains and drag rope. A wider bucket also tends to nod and overshoot when it disengages and the dump rope loads will be higher (which is already higher on the **WRIGHT BUCKET** than on most other buckets). All of the above will have to be discussed with mine engineers and maintenance managers. Thereafter a 6 cubic meter bucket should be build and tested and then changes can be made to the **WRIGHT BUCKET**.

The advantages and disadvantages of a wider bucket can be summarised as follows:

- **Advantages**
    - Shorter filling distance (10 to 15%)
    - Lower energy consumption (20%)
    - No increase in required drag force
- **Disadvantages**
    - Higher stresses
    - Higher dump rope loads
    - Less stability while swinging
    - Nodding and overshooting
    - More wear on drag chains

Specific research will have to be done to address the problems associated with a wider bucket. A comprehensive FEA should be done to determine the weight penalty associated with a wider bucket. The research should also be extended to rigging design. Developing a dynamic model of the bucket after disengagement will result in the ability to easily obtain the forces in the different components, the carry angle of the bucket after disengagement, stability of the bucket and the dumping performance. This will address most of the problems associated with the rigging.

The geometric parameters that should still be investigated are bucket height, the angle of the bucket back to the horizontal and the outwards taper of the sides. The influence of a cut-down back should also be investigated as there exists different opinions on its effect. Tests should also be done for different conditions. In this project the slope of the digging face was flat and at an angle equal to the drag angle, since this was reproducible. This will not always be the case. A possible way to test different conditions would be to prepare a block and measure all performance variables for excavating the whole block. This will result in a design being tested over a range of conditions.

Some more development that needs to be done on rigging design is on quick changeover technology. This will decrease the amount of downtime when a bucket has to be changed and will allow for the design of buckets for a specific purpose, for example a chop-down bucket, rock bucket, sand bucket, etc. The weight of the rigging can also be reduced when the forces are known and a careful fatigue analyses has been done. Research must also be done on ways to decrease the wear on the bucket, without adding weight, which will allow longer cycles between bucket changes. Then there are also room for new concepts on bucket and rigging design. It was observed that operators tend to overdrag the bucket (dragging it even after it has filled) rather than picking it up to far from the fairleads. The reason for this is that the carry angle of the bucket is to low far from the fairleads, resulting in soil being lost from the mouth. In the process however, time is being lost and wear on the bucket is high. If a rigging concept can be developed that will allow the carry angle to be less dependant on the pick-up position it will cause dramatic increases in productivity.

Research should also be conducted on the soil characteristics. Ideally one would like to obtain the bucket design for a given soil, which means the soil should be quantifiable. The important characteristics and ways to obtain them from a site should be included in this study. This is also necessary if a numeric model of the digging process is to be developed in the future. In the authors mind it would be very difficult to develop a model like that, but, if it was possible to model the digging process accurately, it would lead to a revolution in bucket design as numerous designs could be tested for numerous conditions at low costs.

10. REFERENCES

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APPENDIX A

TABLES RELATING TO THE PRE-TEST RANKING OF VARIABLES

Table A1: Determining the influence of bucket width on filling distance and specific digging energy

	filling distance (m)				specific digg. energy					
	STD	hh	UQ	hh	STD	hh	UQ	hh	diff FD	diff SDE
Gravel SL 1	2.55	3	2.04	2	74.44	2	61.32	2	0.51	13.12
Gravel SL 2	2.5	4	2.07	3	79.24	4	66.2	3	0.43	13.04
Gravel LL 1	3.1	2	2.36	1	92.84	2	72.8	1	0.74	20.04
Gravel LL 2	2.74	3	2.37	2	89.12	2	77.76	2	0.37	11.36
Gravel SH 1	2.52	3	2.04	2	72.64	2	62.88	2	0.48	9.76
Gravel SH 2	2.8	4	2.01	3	84.2	3	67.64	4	0.79	16.56
Deco SL 1	2.23	3	1.84	2	78.9	2	66.48	1	0.39	12.42
Deco SL 2	2.29	3	1.77	3	99.8	3	65.88	2	0.52	33.92
Deco LL 1	2.63	1			106.8	1				
Deco LL 2	2.49	2	2.15	1	122.94	2	90.54	1	0.34	32.4
Deco SH 1	2.53	3	2.03	2	99.34	2	69.04	1	0.5	30.3
Deco SH 2	2.66	4	1.97	2	109.19	3	83.59	2	0.69	25.6
average	2.587		2.06		92.454		71.285		0.5236	19.865
standard dev	0.23		0.18		15.584		9.1259		0.1523	9.1069

Table A2: Determining the influence of teeth length on filling distance and specific digging energy

	filling distance (m)				specific digg. energy					
	short	hh	long	hh	short	hh	long	hh	diff FD	diff SDE
STD Gravel L1	2.55	3	3.1	2	74.44	2	92.84	2	-0.55	-18.4
STD Gravel L2	2.5	4	2.74	3	79.24	4	89.12	2	-0.24	-9.88
UQ Gravel L1	2.04	2	2.36	1	61.32	2	72.8	1	-0.32	-11.48
UQ Gravel L2	2.07	3	2.37	2	66.2	3	[illegible].76	2	-0.3	-11.56
STD Deco L1	2.23	3	2.63	1	78.9	2	106.8	1	-0.4	-27.9
STD Deco L2	2.29	3	2.49	2	99.8	3	122.94	2	-0.2	-23.14
UQ Deco L1	1.84	2			66.48	1				
UQ Deco L2	1.77	3	2.15	1	65.88	2	90.54	1	-0.38	-24.66
average	2.161		2.55		74.033		93.257		-0.341	-18.15
standard dev	0.284		0.31		12.314		17.068		0.1161	7.2883

Table A3: **Determining the influence of digging material on filling distance and specific digging energy**

	filling distance (m)				specific digg. energy					
	Gravel	hh	Deco	hh	Gravel	hh	Deco	hh	diff FD	diff SDE
STD SL 1	2.55	3	2.23	3	74.44	2	78.9	2	0.32	-4.46
STD SL 2	2.5	4	2.29	3	79.24	4	99.8	3	0.21	-20.56
STD LL 1	3.1	2	2.63	1	92.84	2	106.8	1	0.47	-13.96
STD LL 2	2.74	3	2.49	2	89.12	2	122.94	2	0.25	-33.82
STD SH 1	2.52	3	2.53	3	72.64	2	99.34	2	-0.01	-26.7
STD SH 2	2.8	4	2.66	4	84.2	3	109.19	3	0.14	-24.99
UQ SL 1	2.04	2	1.84	2	61.32	2	66.48	1	0.2	-5.16
UQ SL 2	2.07	3	1.77	3	66.2	3	65.88	2	0.3	0.32
UQ LL 1	2.36	1			72.8	1				
UQ LL 2	2.37	2	2.15	1	77.76	2	90.54	1	0.22	-12.78
UQ SH 1	2.04	2	2.03	2	62.88	2	69.04	1	0.01	-6.16
UQ SH 2	2.01	3	1.97	2	67.64	4	83.59	2	0.04	-15.95
average	2.425		2.24		75.09		90.227		0.1955	-14.93
standard dev	0.347		0.31		10.019		19.124		0.1447	10.721

Table A4: **Determining the influence of angle of attack on filling distance and specific digging energy**

	filling distance (m)				specific digg. energy					
	low	hh	high	hh	low	hh	high	hh	diff FD	diff SDE
STD Gravel S1	2.55	3	2.52	3	74.44	2	72.64	2	0.03	1.8
STD Gravel S2	2.5	4	2.8	4	79.24	4	84.2	3	-0.3	-4.96
UQ Gravel S1	2.04	2	2.04	2	61.32	2	62.88	2	0	-1.56
UQ Gravel S2	2.07	3	2.01	3	66.2	3	67.64	3	0.06	-1.44
STD Deco S1	2.23	3	2.53	3	78.9	2	99.34	2	-0.3	-20.44
STD Deco S2	2.29	3	2.66	4	99.8	3	109.19	3	-0.37	-9.39
UQ Deco S1	1.84	2	2.03	2	66.48	1	69.05	1	-0.19	-2.57
UQ Deco S2	1.77	3	1.97	2	65.88	2	83.59	2	-0.2	-17.71
average	2.161		2.32		74.033		81.066		-0.159	-7.034
std dev	0.284		0.34		12.314		16.333		0.1673	8.1308

Table A5: **Determining the influence of position of COG on filling distance and specific digging energy**

	filling distance (m)				specific digg. energy					
	B1	hh	B2	hh	B1	hh	B2	hh	diff FD	diff SDE
STD Gravel SL	2.55	3	2.5	4	74.44	2	79.24	4	0.05	-4.8
STD Gravel LL	3.1	2	2.74	3	92.84	2	89.12	2	0.36	3.72
STD Gravel SH	2.52	3	2.8	4	72.64	2	84.2	3	-0.28	-11.56
UQ Gravel SL	2.04	2	2.07	3	61.32	2	66.2	3	-0.03	-4.88
UQ Gravel LL	2.36	1	2.37	2	72.8	1	77.76	2	-0.01	-4.96
UQ Gravel SH	2.04	2	2.01	3	62.88	2	67.64	4	0.03	-4.76
STD Deco SL	2.23	3	2.29	3	78.9	2	99.8	3	-0.06	-20.9
STD Deco LL	2.63	1	2.49	2	106.8	1	122.94	2	0.14	-16.14
STD Deco SH	2.53	3	2.66	4	99.34	2	109.19	3	-0.13	-9.85
UQ Deco SL	1.84	2	1.77	3	66.48	1	65.88	2	0.07	0.6
UQ Deco LL			2.15	1			90.54	1		
UQ Deco SH	2.03	2	1.97	2	69.04	1	83.59	2	0.06	-14.55
average	2.352		2.32		77.953		86.342		0.0182	-8.007
standard dev	0.362		0.33		15.146		17.525		0.1604	7.3668

Table A6: Comparison between hitch heights for minimum filling distance and maximum engagement rate

	FDmin	HH	ERmax	HH	HH same
STD Gravel SL1	2.55	3	11.75	3	yes
STD Gravel SL2	2.5	4	14.98	4	yes
STD Gravel LL1	3.1	2	17.98	2	yes
STD Gravel LL2	2.74	3	17.14	3	yes
STD Gravel SH1	2.52	3	11.32	3	yes
STD Gravel SH2	2.8	4	10.9	4	yes
UQ Gravel SL1	2.04	2	11.32	2	yes
UQ Gravel SL2	2.07	3	14.98	3	yes
UQ Gravel LL1	2.36	1	10.13	1	yes
UQ Gravel LL2	2.37	2	18.89	2	yes
UQ Gravel SH1	2.04	2	12.21	2	yes
UQ Gravel SH2	2.01	3	17.98	3	yes
STD Deco SL1	2.23	3	16.68	3	yes
STD Deco SL2	2.29	3	25.54	4	no
STD Deco LL1	2.63	1	19.86	1	yes
STD Deco LL2	2.49	2	32.27	2	yes
STD Deco SH1	2.53	3	18.52	3	yes
STD Deco SH2	2.66	4	17.89	4	yes
UQ Deco SL1	1.84	2	30.98	2	yes
UQ Deco SL2	1.77	3	27.54	2	no
UQ Deco LL1					
UQ Deco LL2	2.15	1	38.36	1	yes
UQ Deco SH1	2.03	2	13.97	2	yes
UQ Deco SH2	1.97	2	25.54	2	yes

APPENDIX B

FIGURES RELATING TO OPTIMUM CARRY ANGLES

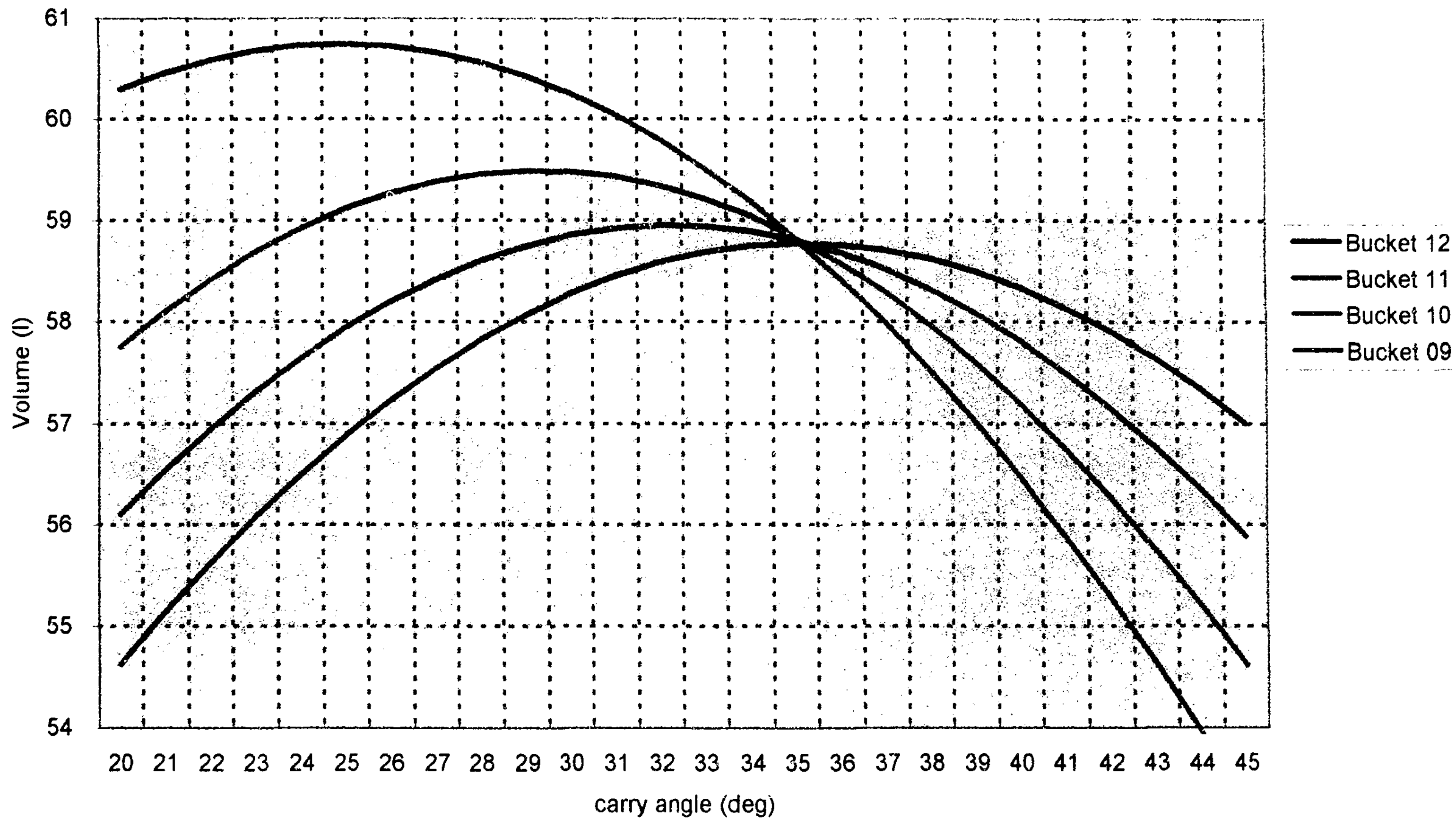

Figure B1: Payload volume as a function of carry angle for the four test buckets as calculated with a 2D theoretical model for an angle of repose of 38 degrees

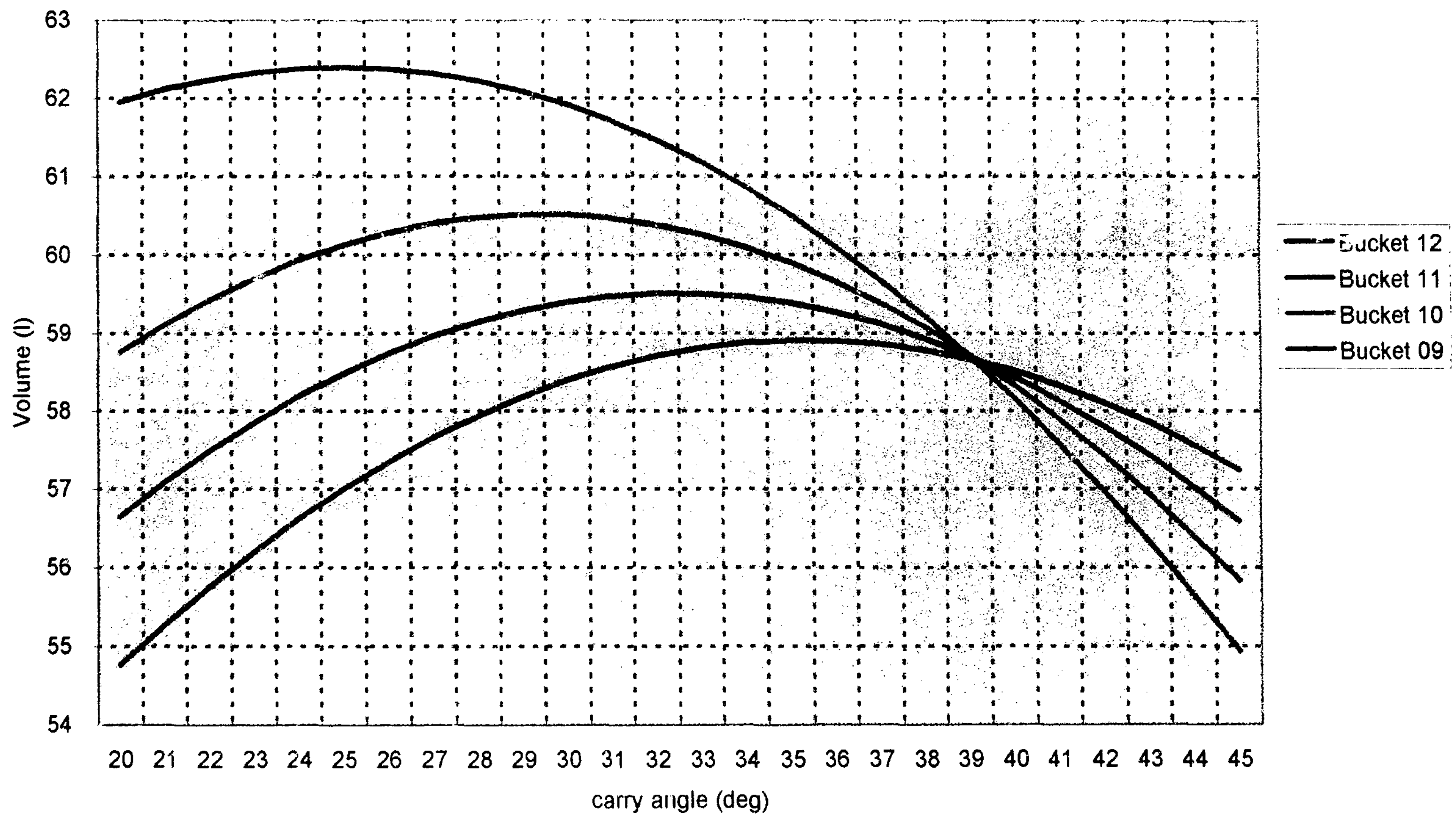

Figure B2: Payload volume as a function of carry angle for the four test buckets as calculated with a 2D theoretical model for an angle of repose of 42 degrees

B3

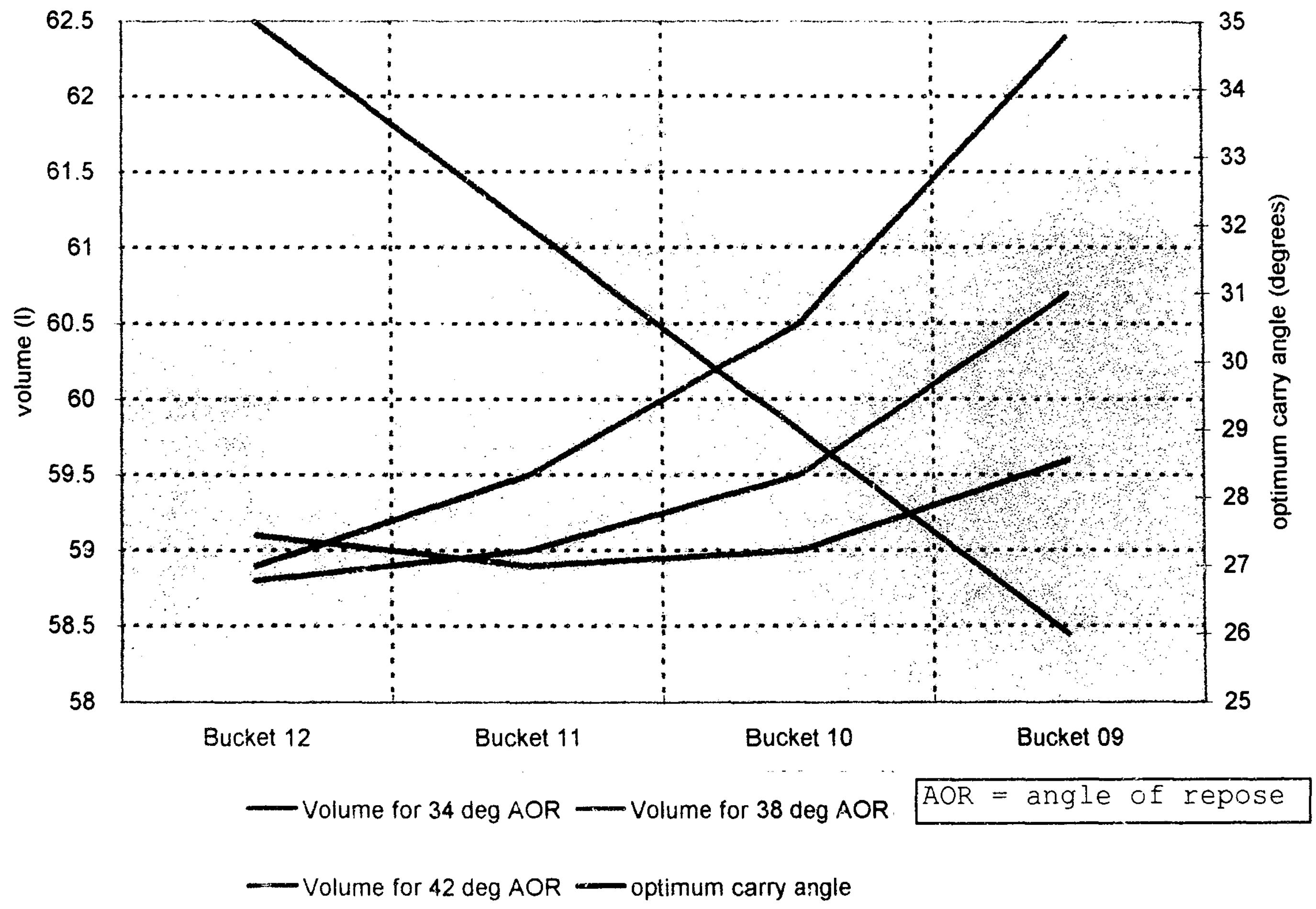

Figure B3: Maximum payload and optimum carry angle for the four buckets for different angles of repose

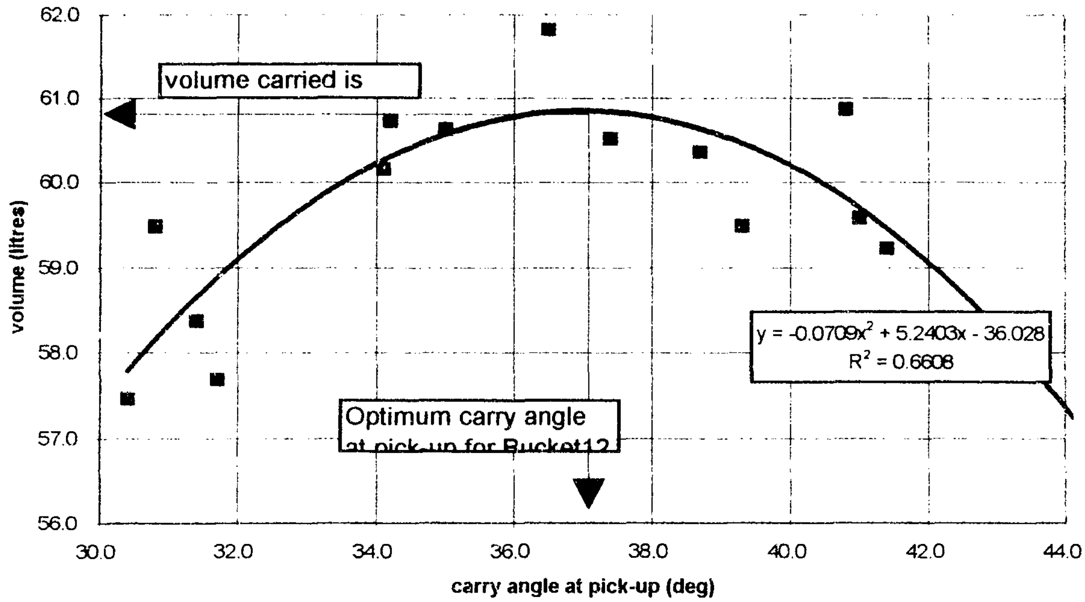

Figure B4: Optimum carry angle at pick-up for Test Bucket 12 in crusher run with angle of repose between 42 and 48 degrees

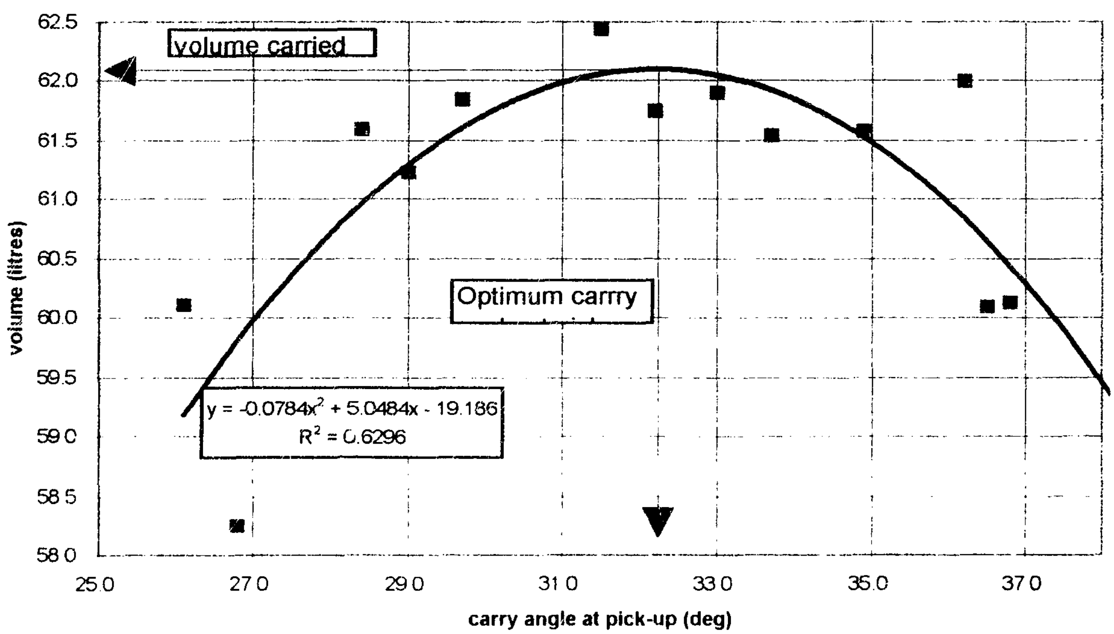

Figure B5: Optimum carry angle at pick-up for Test Bucket 11 in crusher run with angle of repose between 42 and 48 degrees

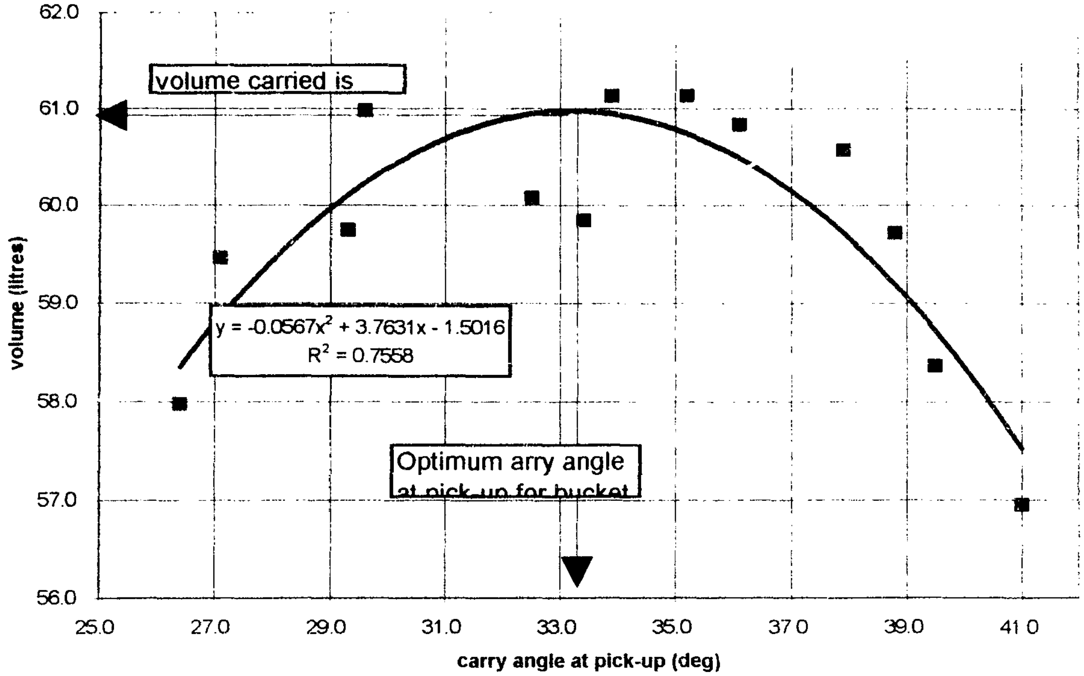

Figure B6: Optimimum carry angle at pick-up for Test Bucket 12 in crusher run with angle of repose between 42 and 48 degrees

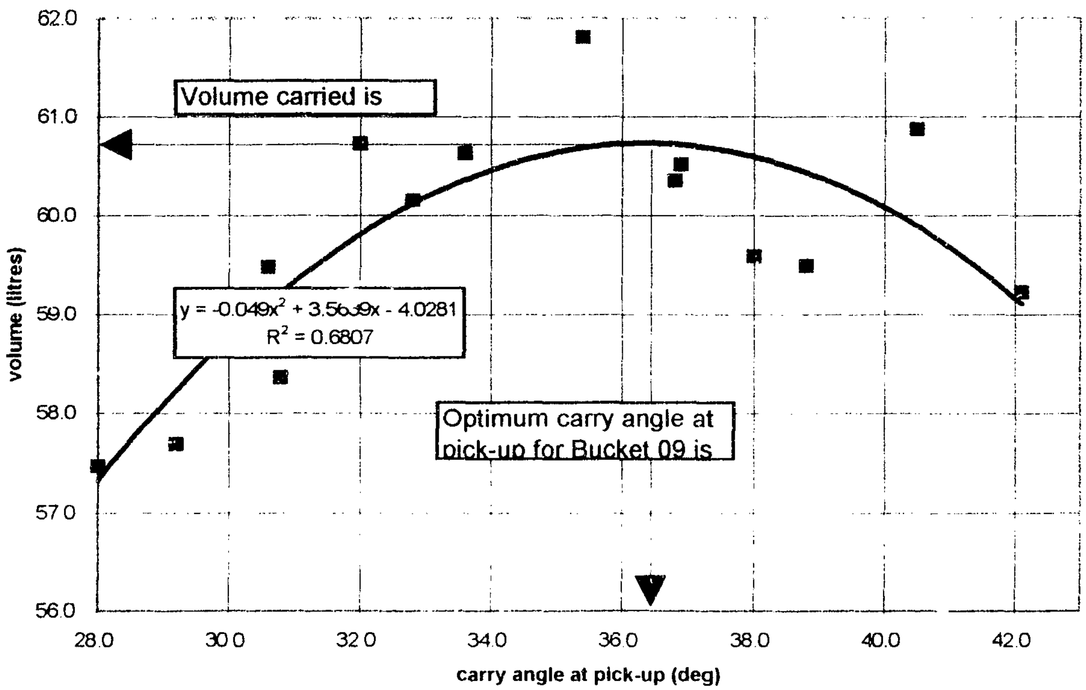

Figure B7: Optimum carry angle at pick-up for Test Bucket 12 in crusher run with angle of repose between 42 and 48 degrees

APPENDIX C

FIGURES RELATING TO THE TEST DATA

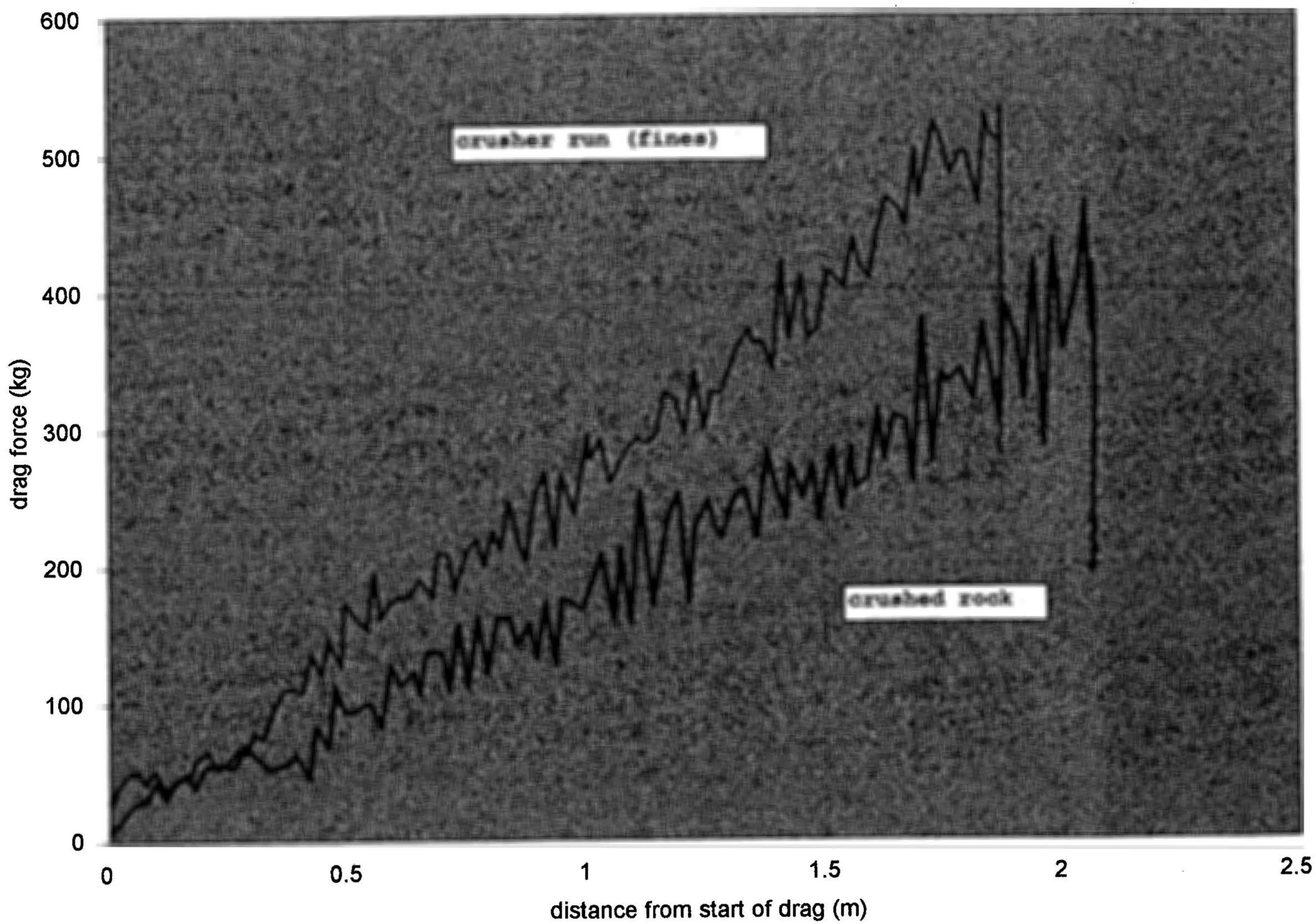

Figure C1: Comparison between Drag Force and Distance in crushed rock and crusher run for tests RK106833 and CR106833

C2

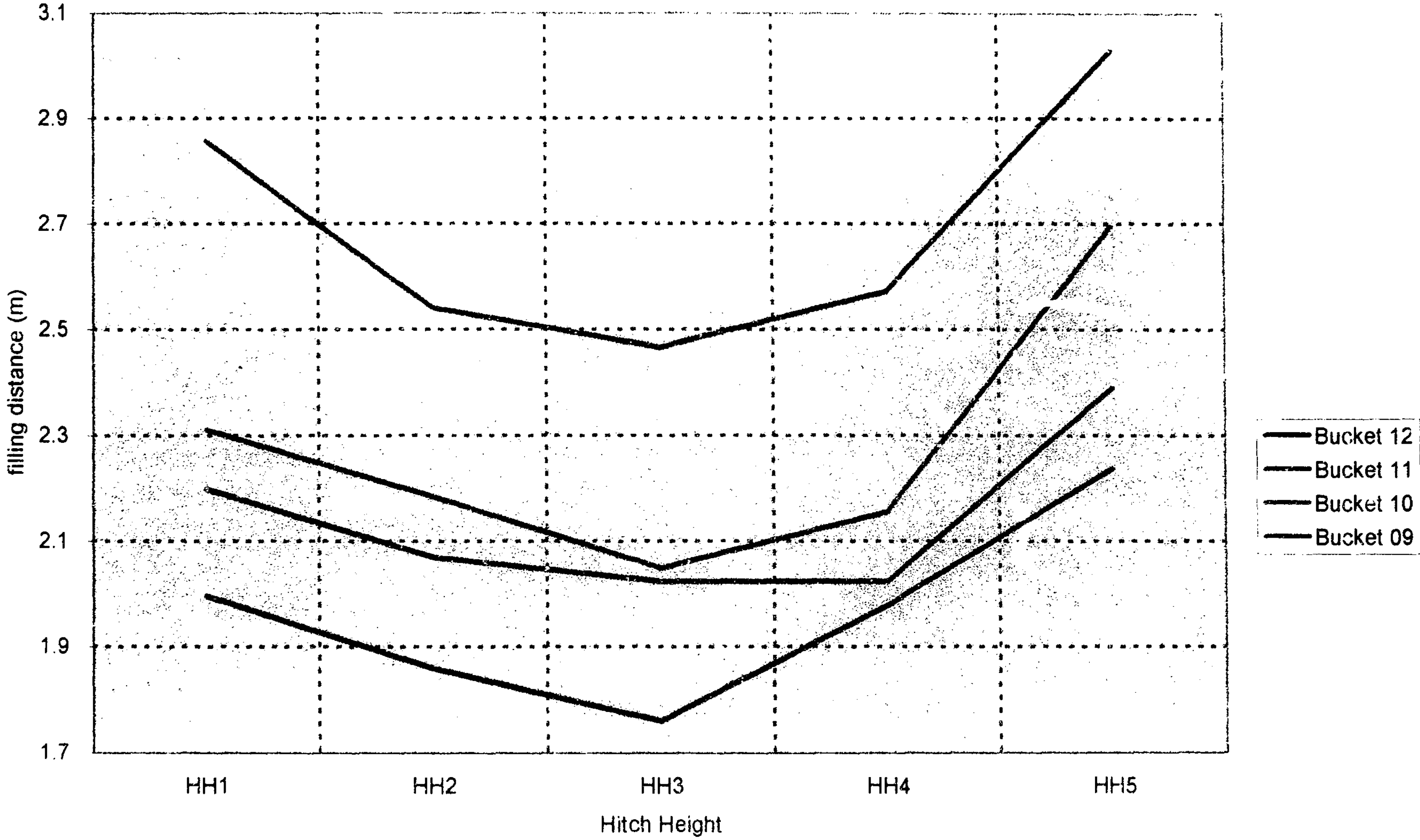

Figure C2: Filling Distance as a function of Hitch Height for the four test buckets in crushed rock.

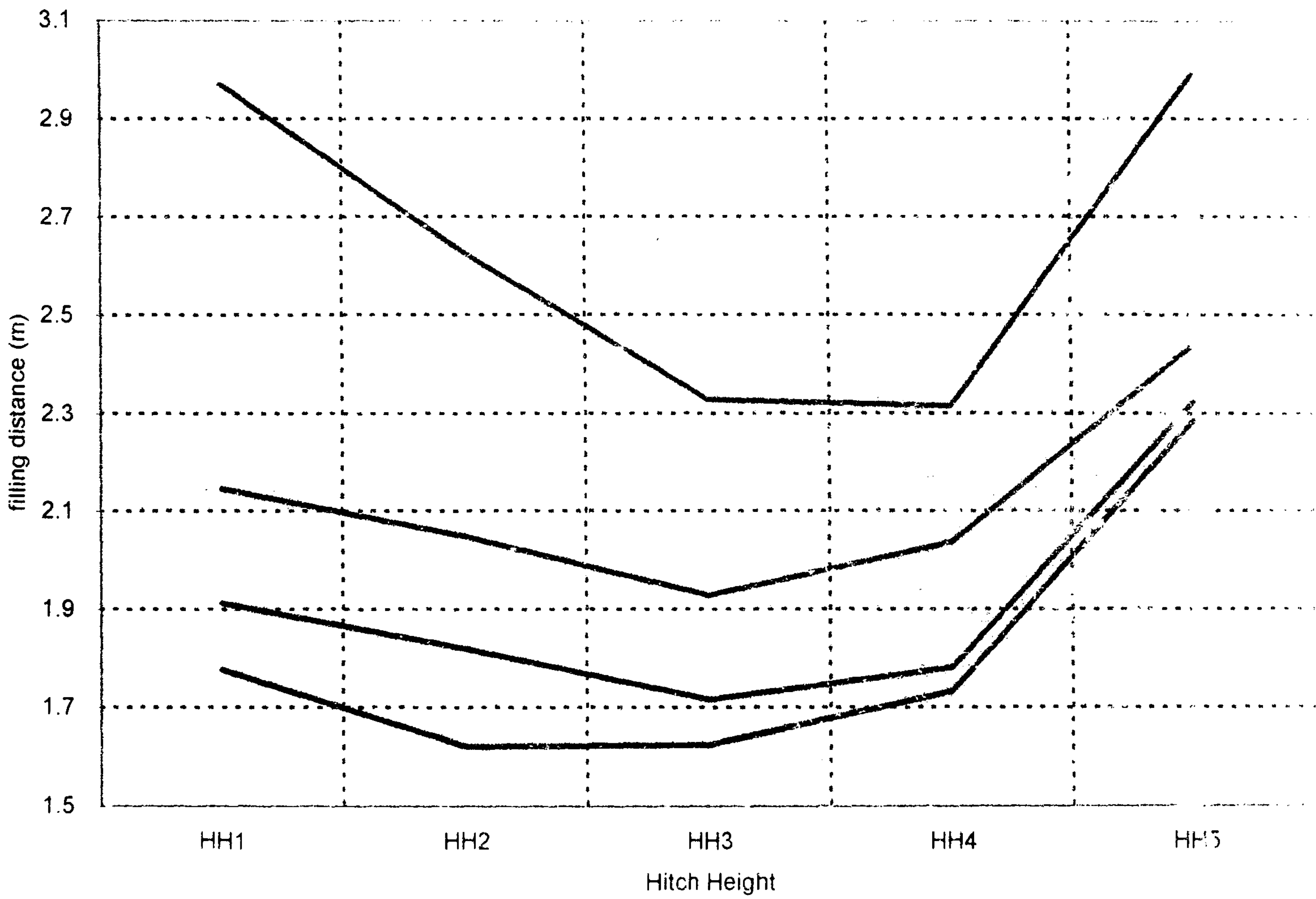

Figure C3: Filling Distance as a function of Hitch Height for the four test buckets in crusher run

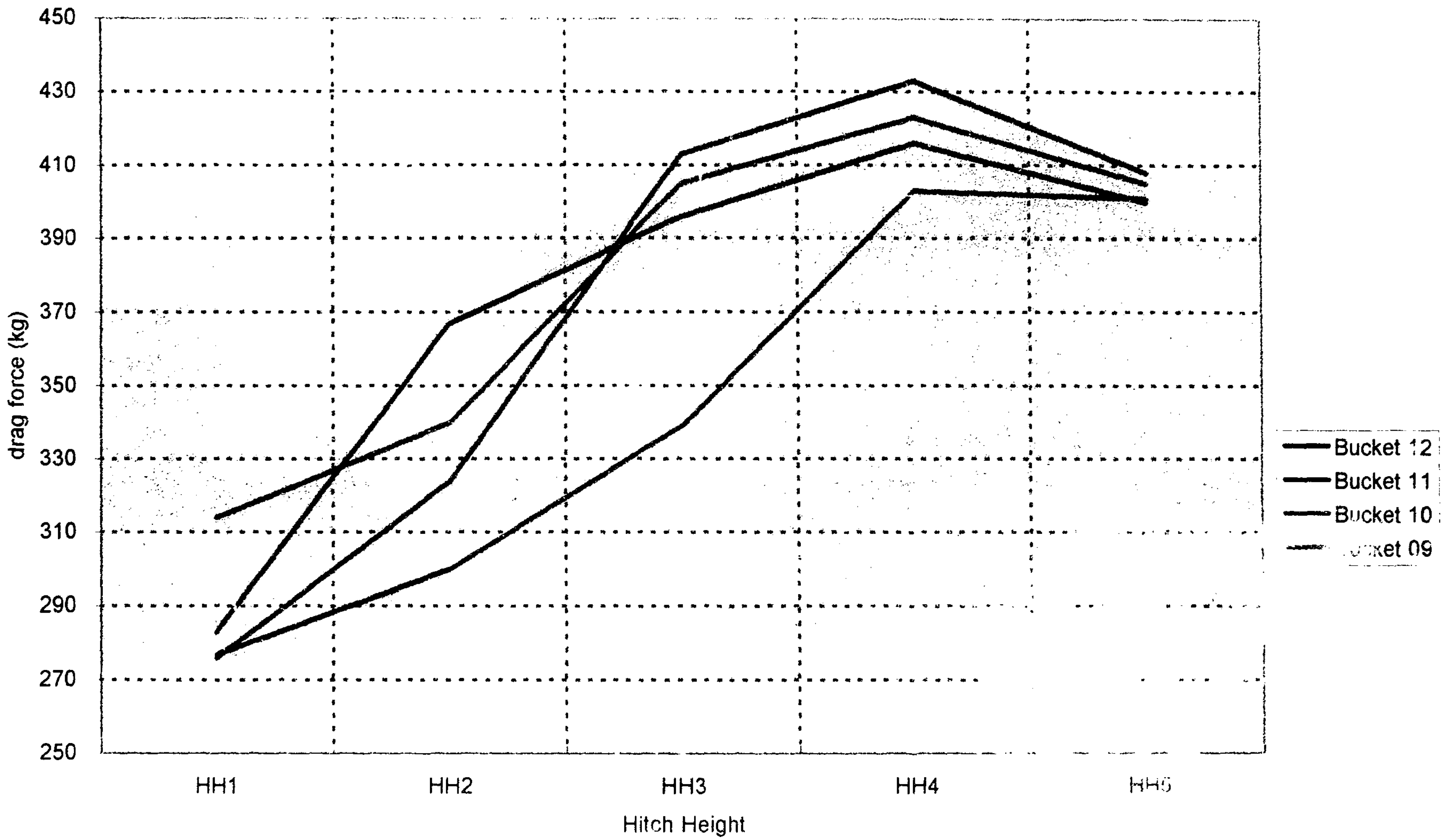

Figure C4: Max Drag Force as a function of Hitch Height for the four test buckets in crushed rock.

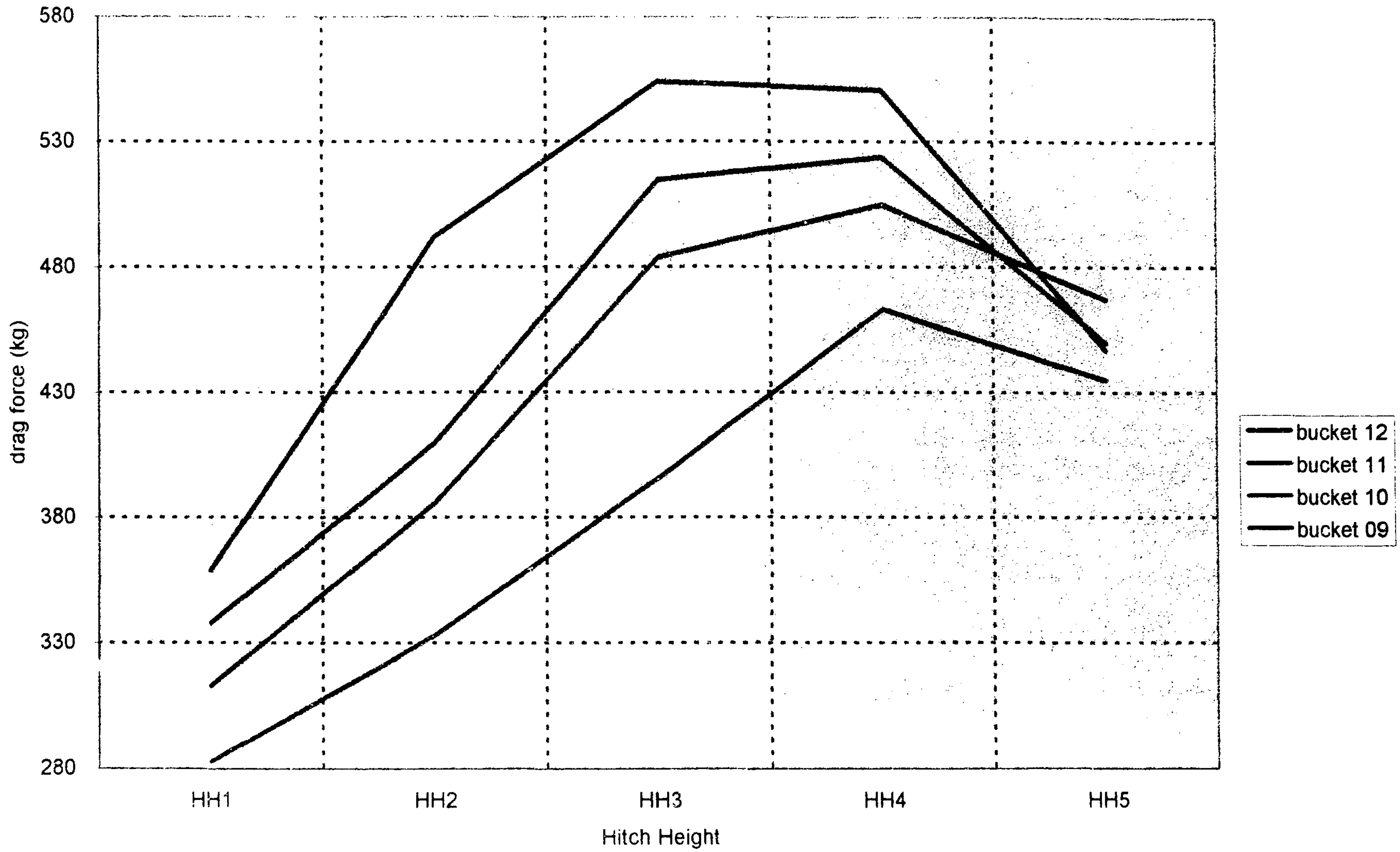

Figure C5: Max Drag Force as a function of Hitch Height for the four test buckets in crusher run.

C5

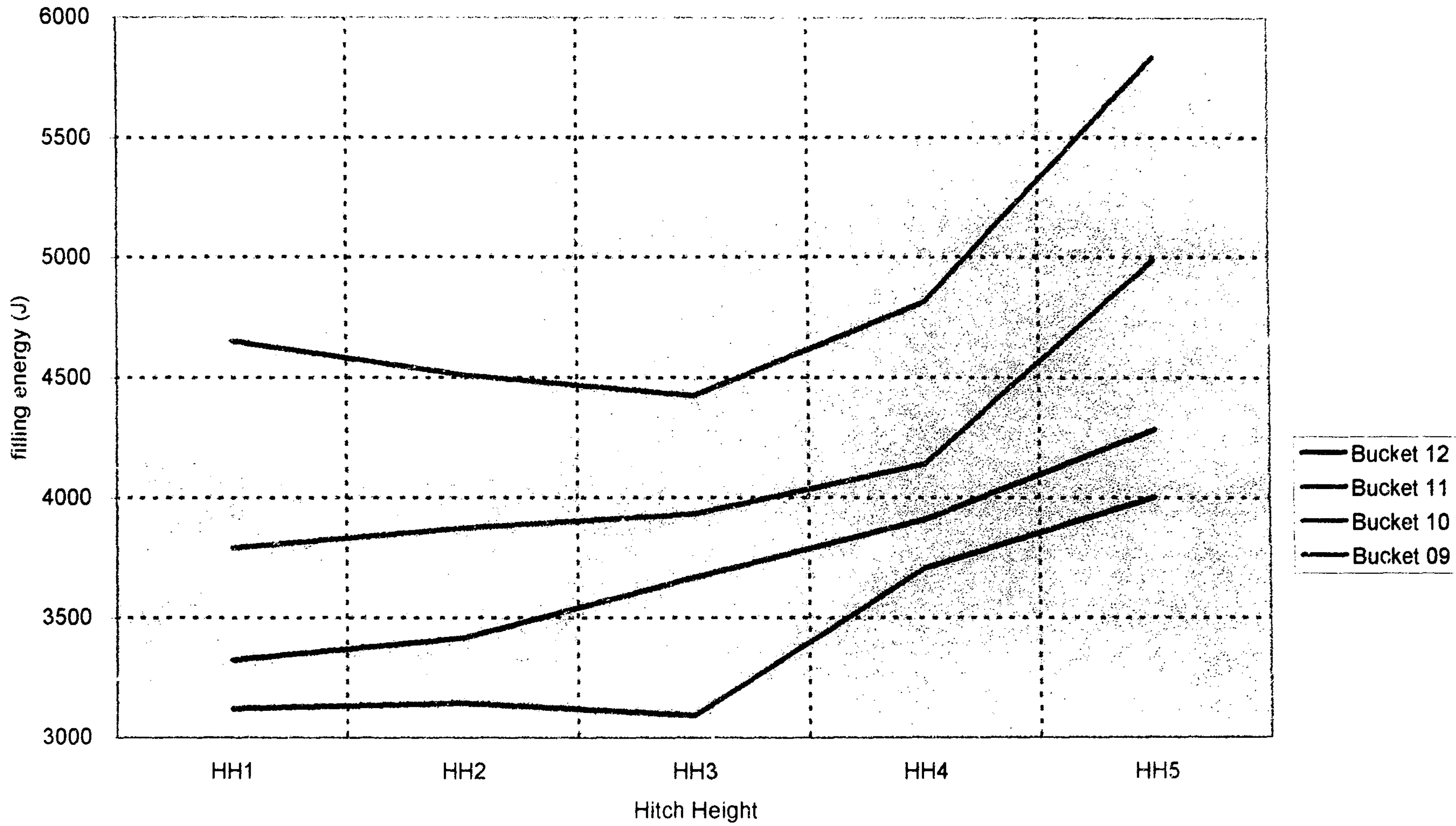

Figure C6: Filling Energy as a function of Hitch Height for the four test buckets in crushed rock.

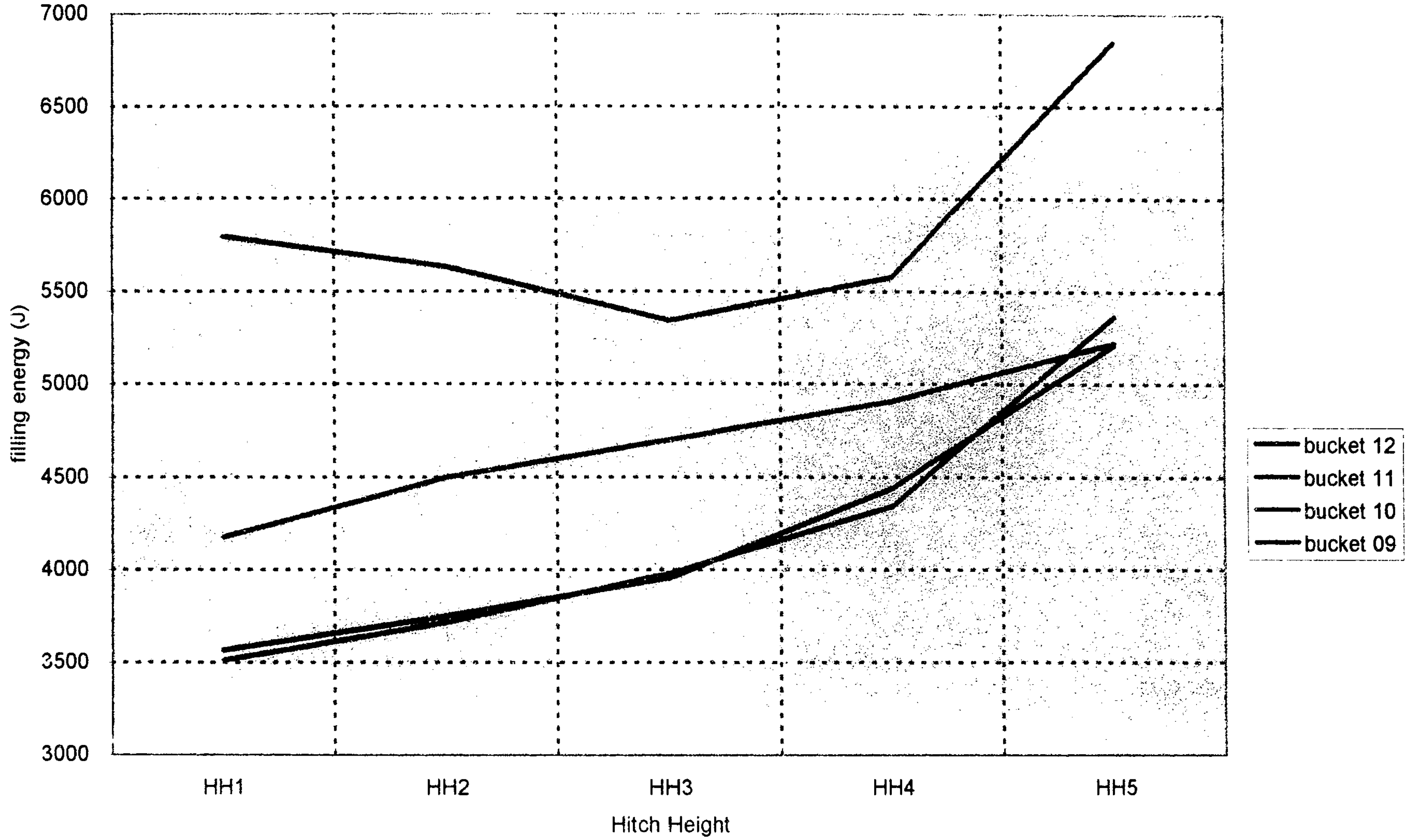

Figure C7: Filling Energy as a function of Hitch Height for all four test buckets in crusher run.

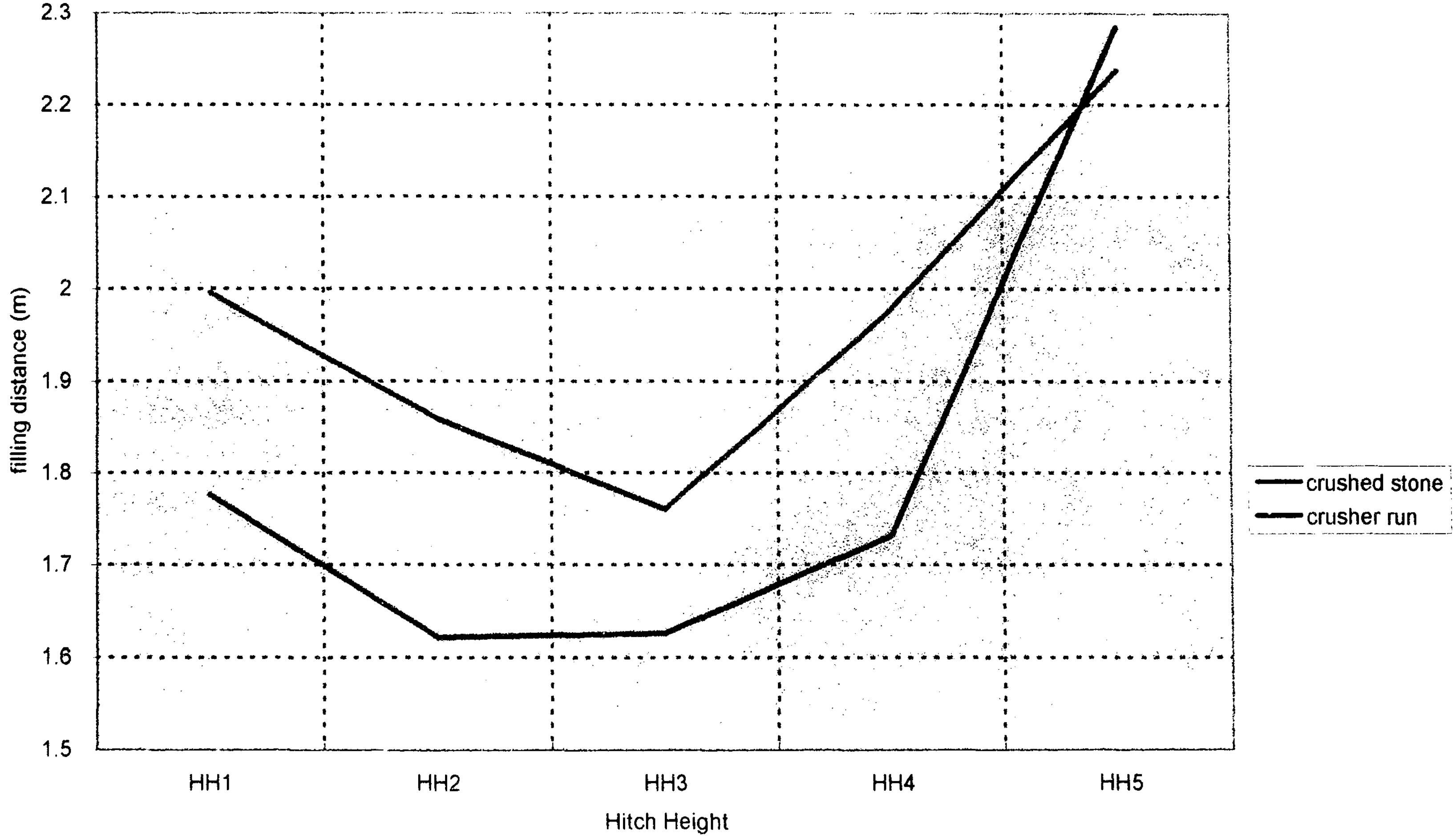

Figure C8: Filling Distance as a function of Hitch Height for bucket 12 in two spoils.

C8

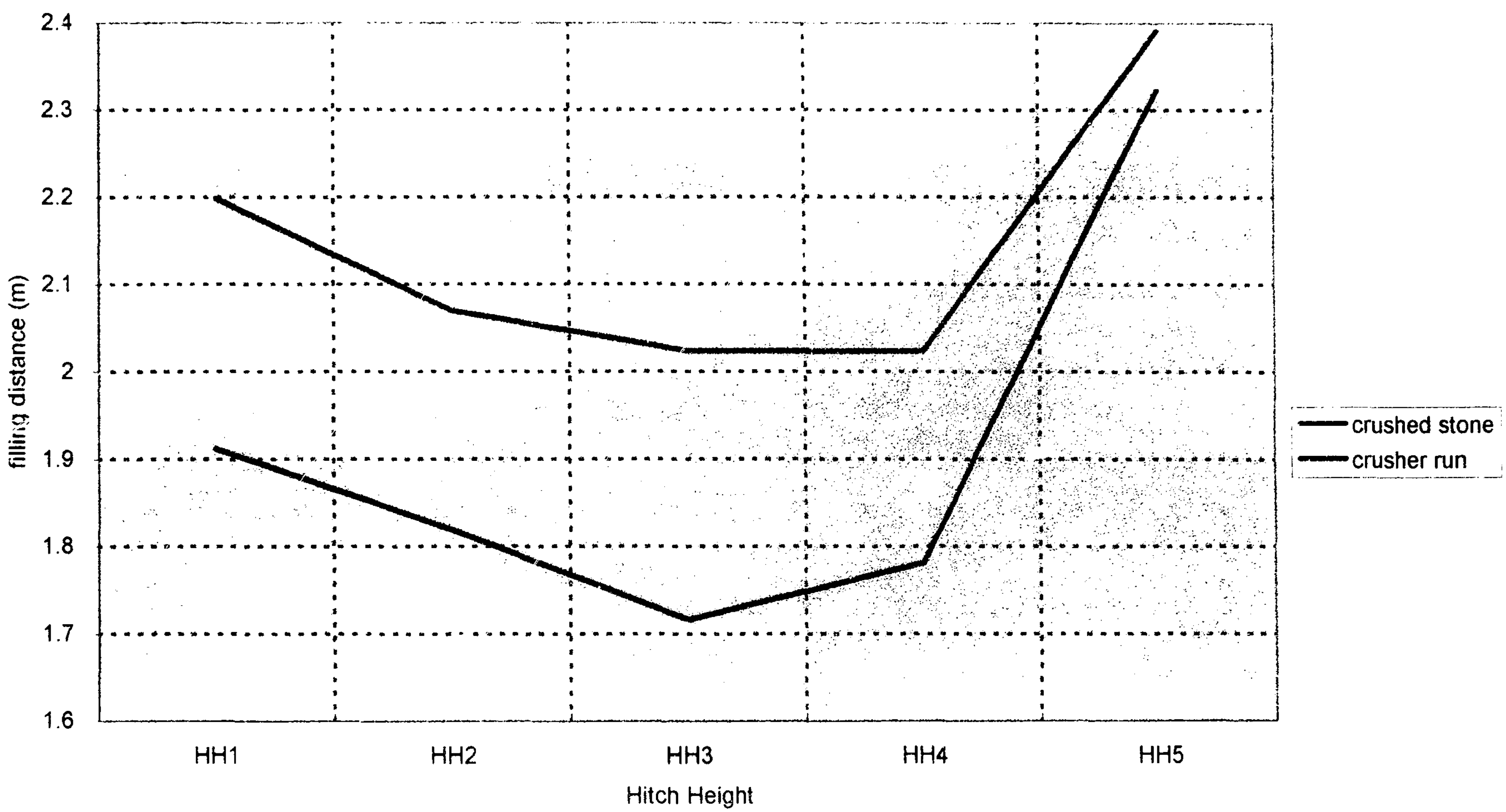

Figure C9: Filling Distance as a function of Hitch Height for bucket 11 in two spoils.

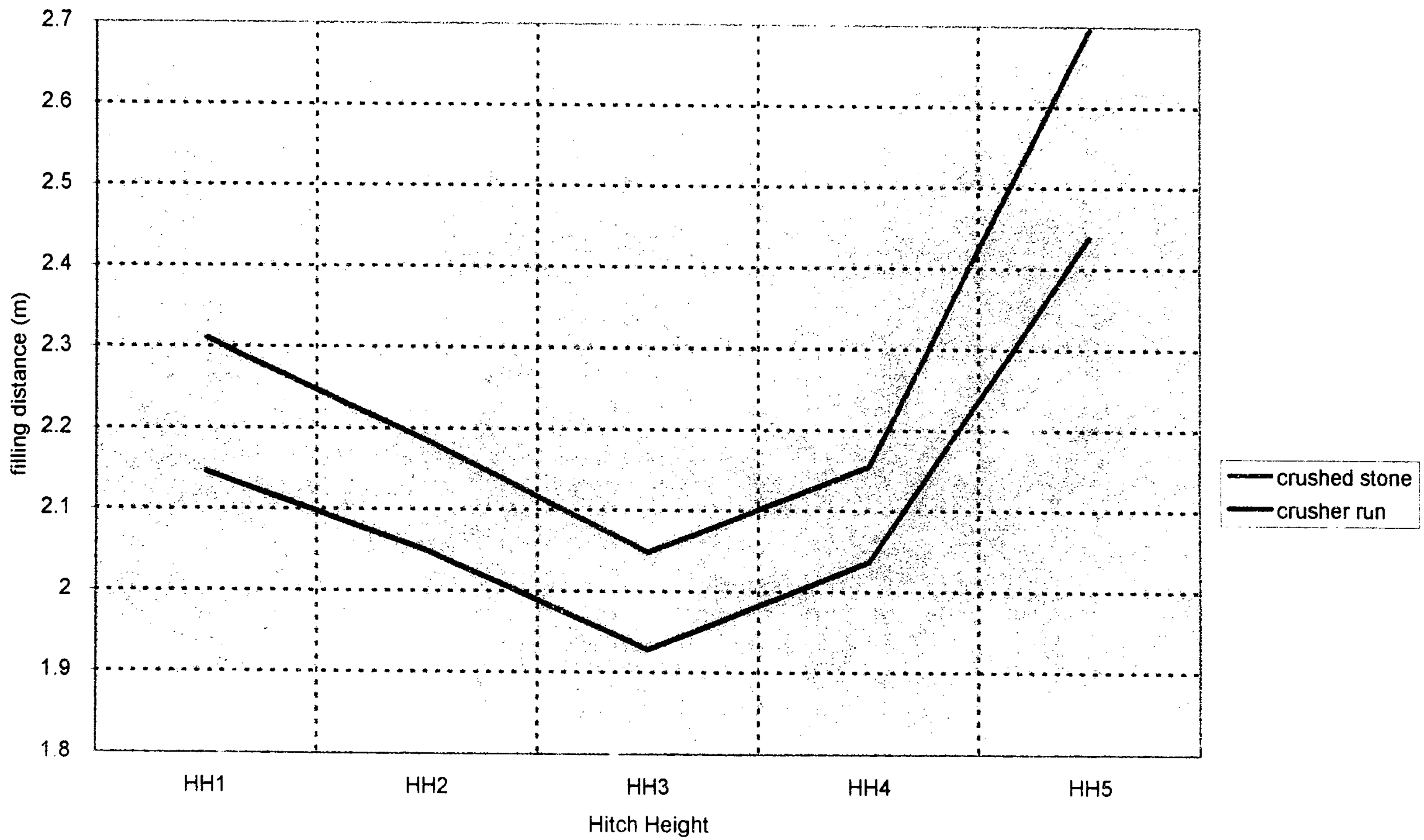

Figure C10: Filling Distance as a function of Hitch Height for bucket 10 in two spoils.

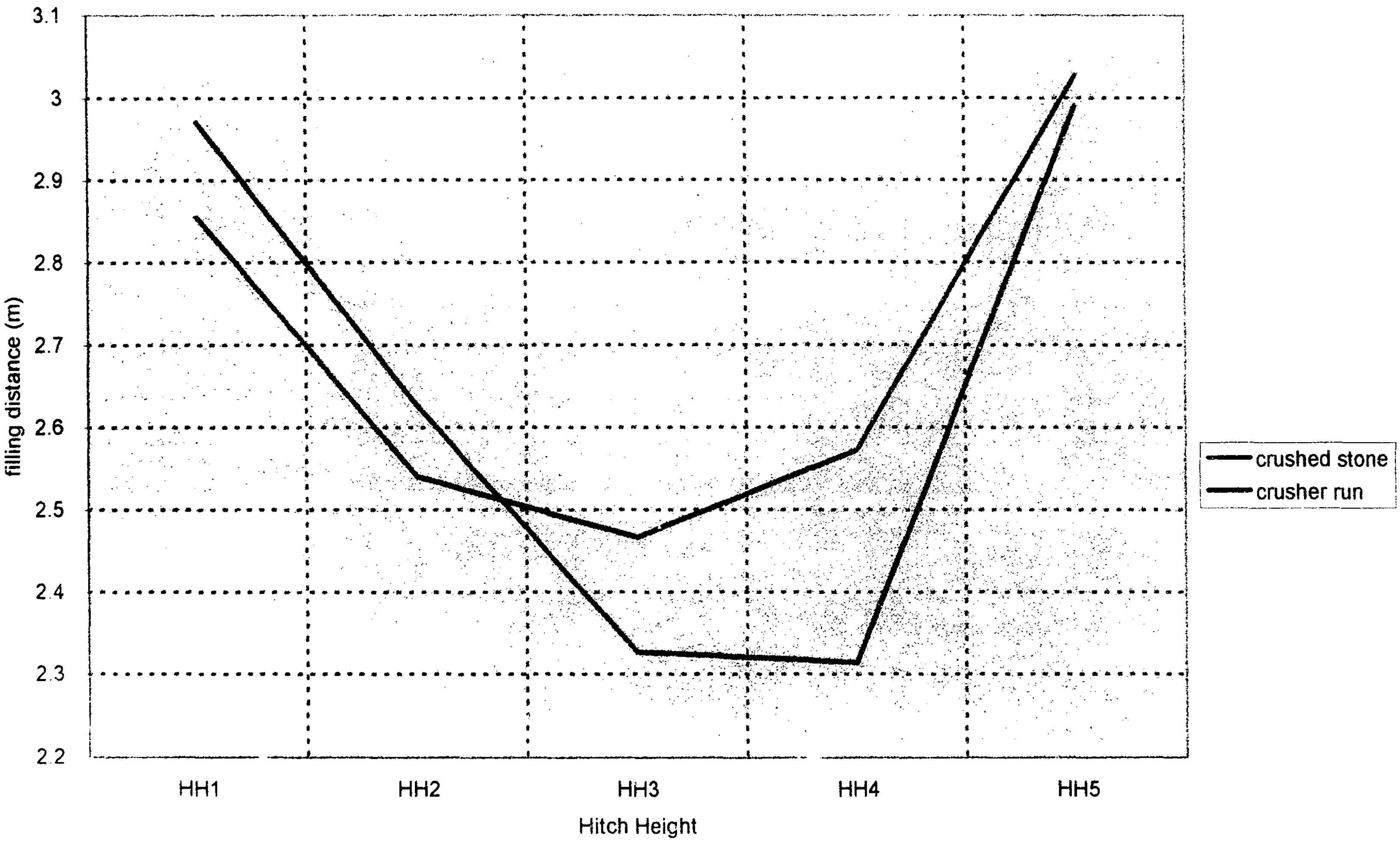

Figure C11: Filling Distance as a function of Hitch Height for bucket 09 in two spoils.

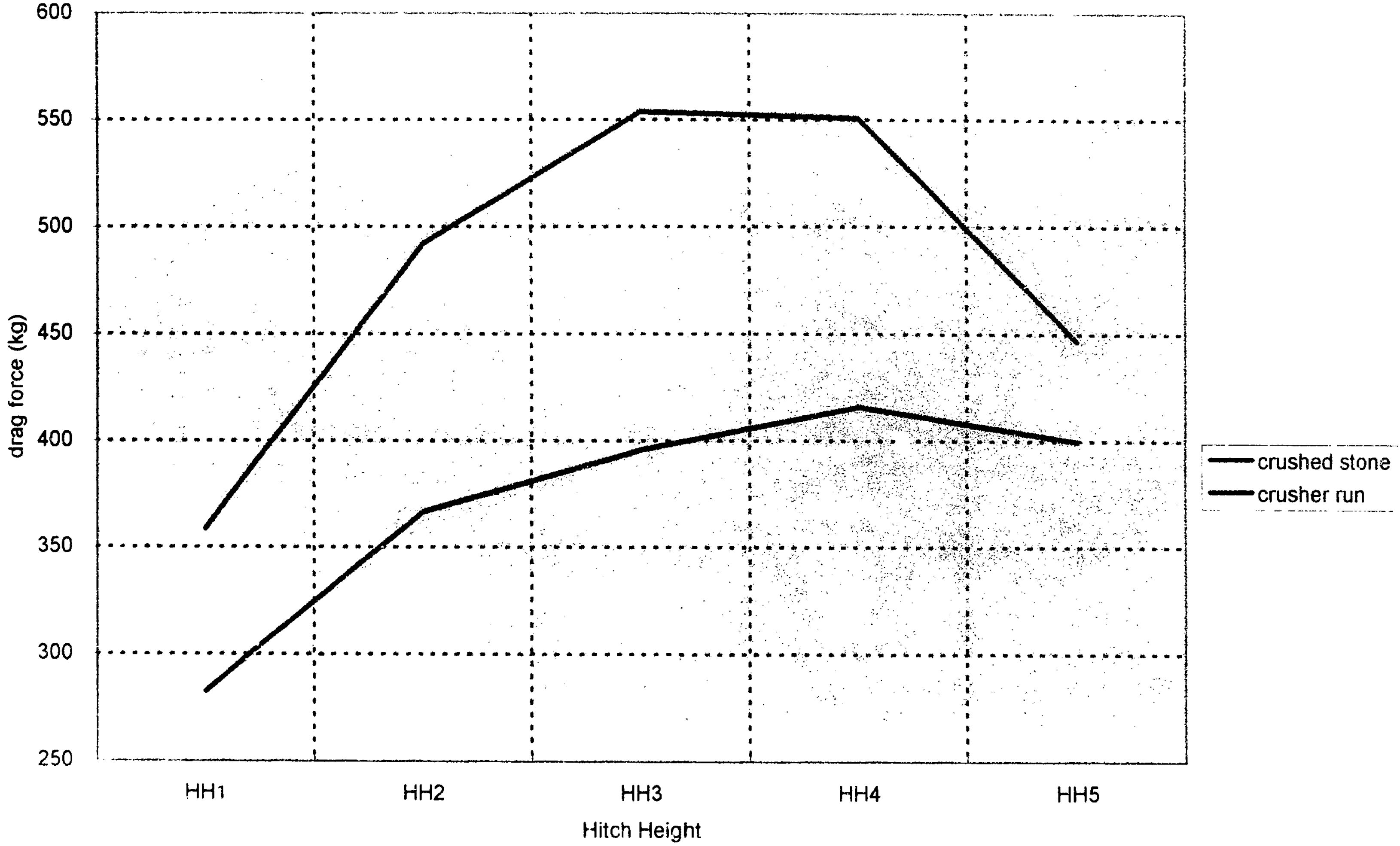

Figure C12: Max Drag Force as a function of Hitch Height for bucket 12 in two spoils.

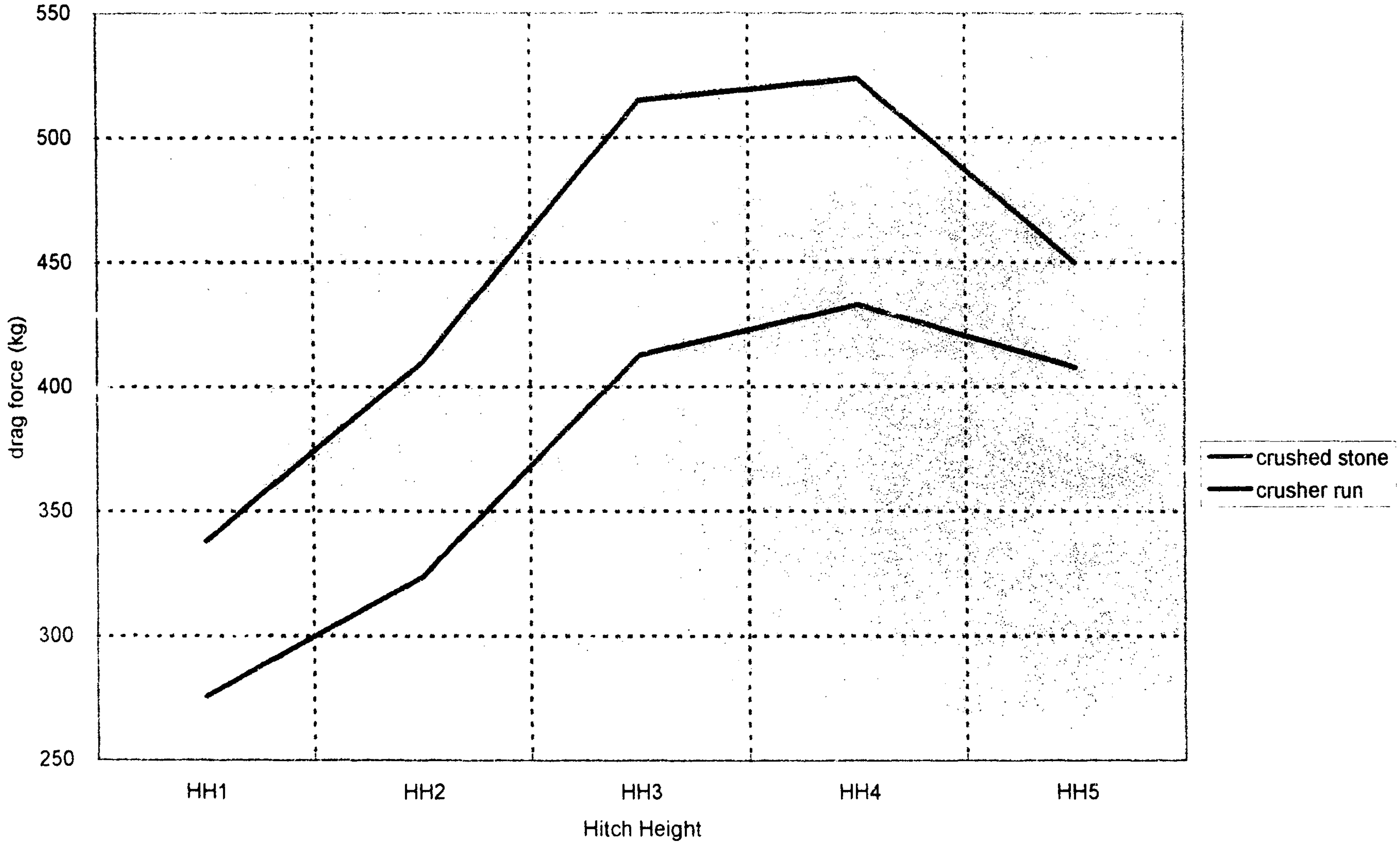

Figure C13: Max Drag Force as a function of Hitch Height for bucket 11 in two spoils.

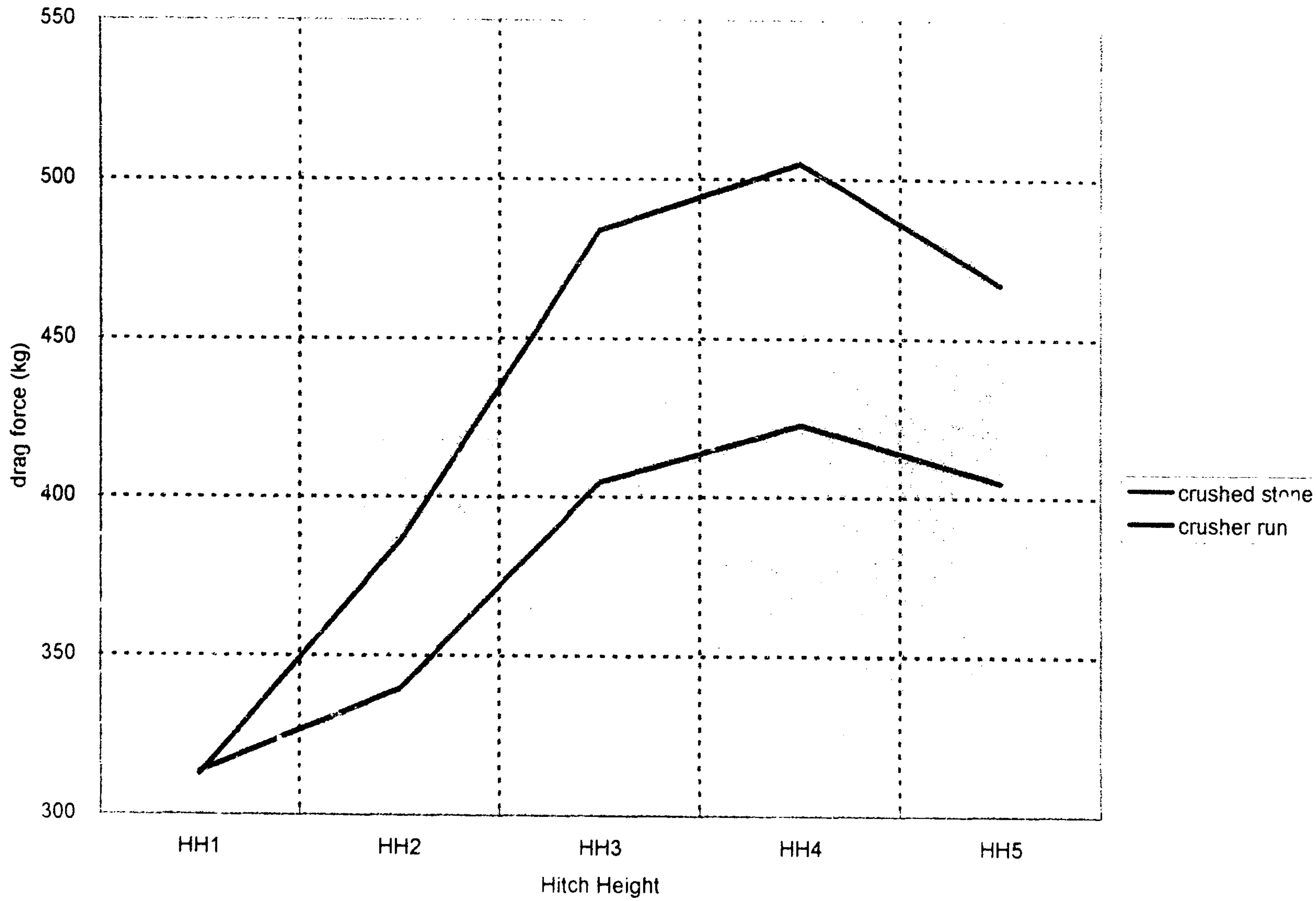

Figure C14: Max Drag Force as a function of Hitch Height for bucket 10 in two spoils.

C14

C15

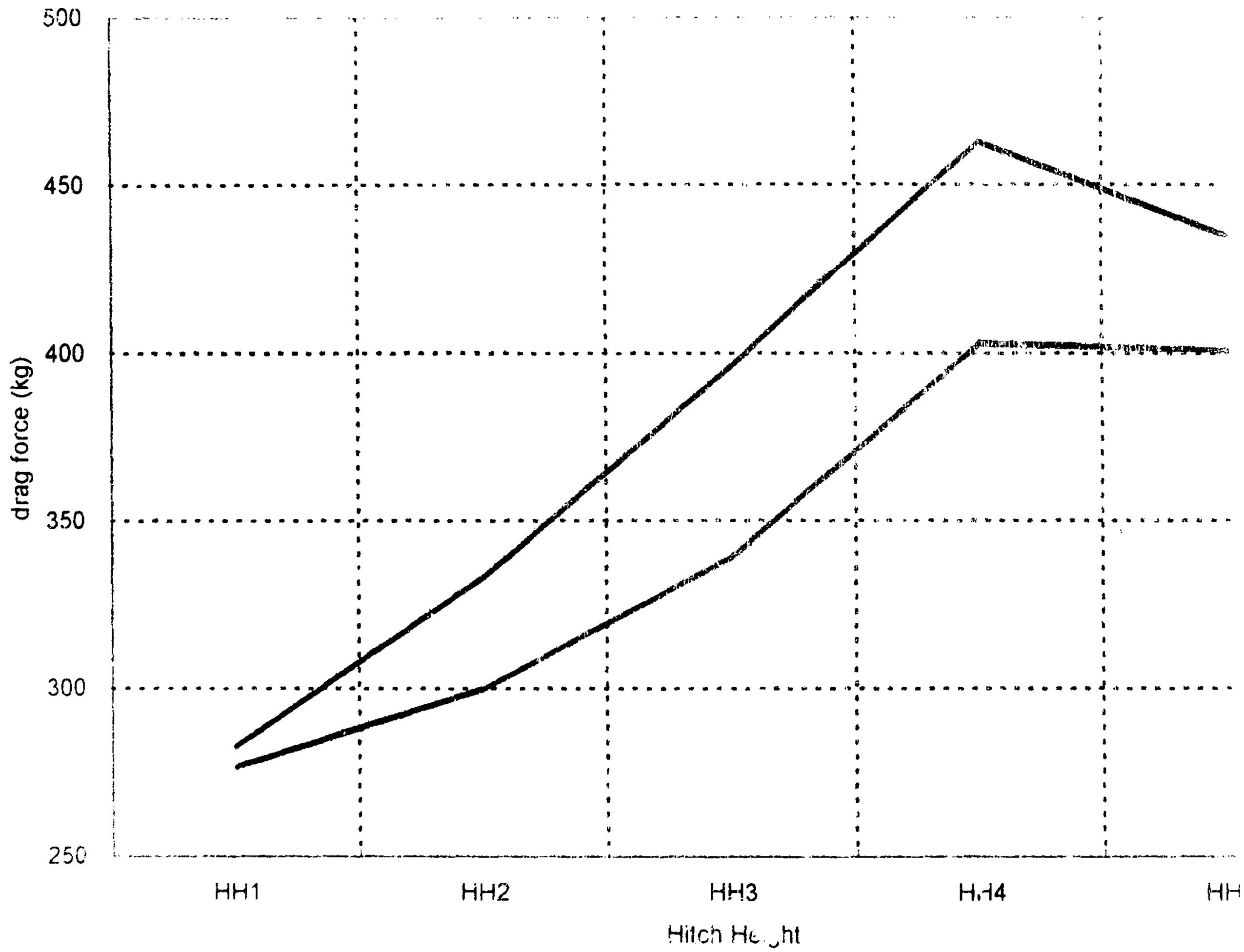

Figure C15: Max Drag Force as a function of Hitch Height for bucket 09 in two spoils.

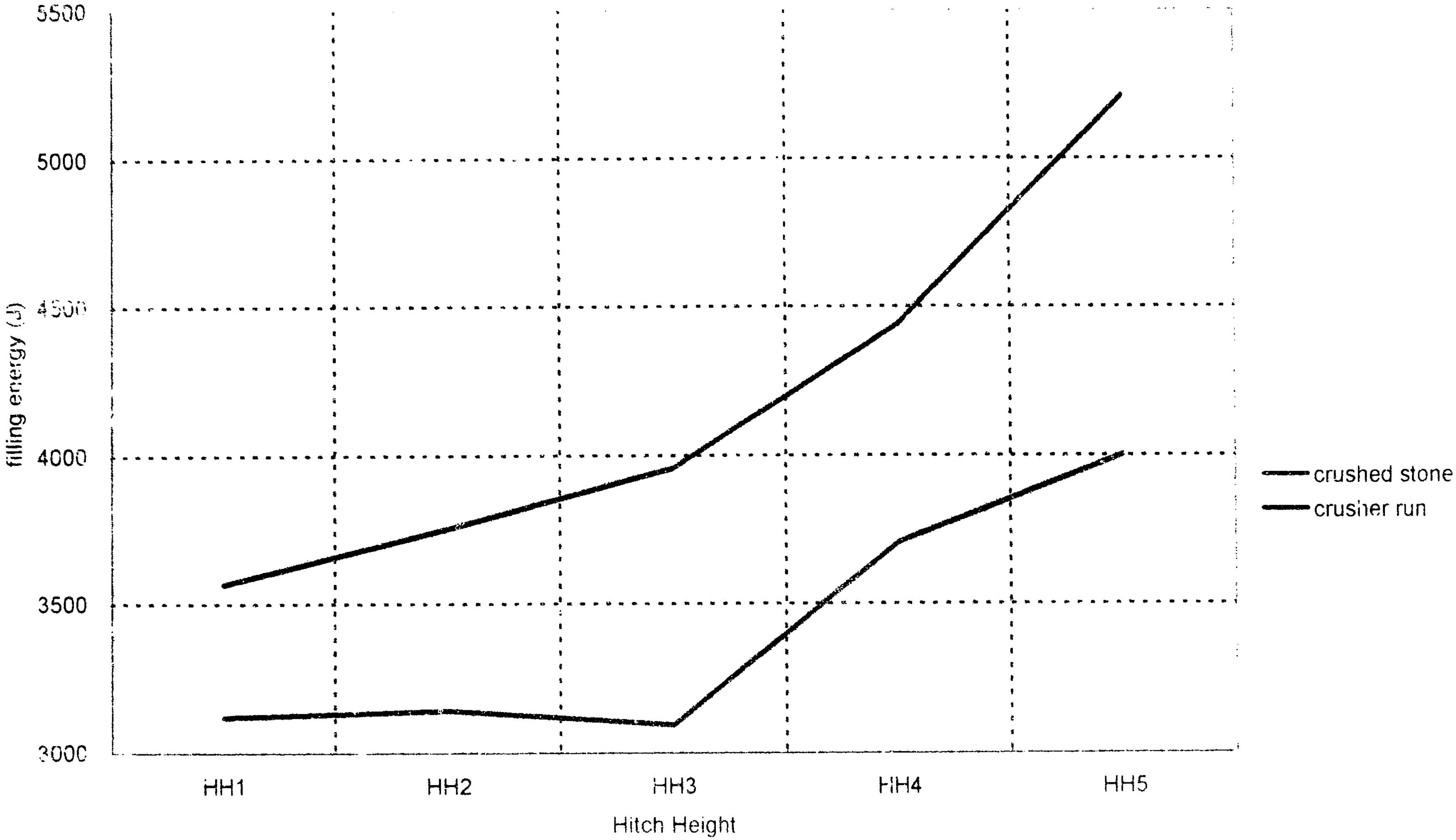

Figure C16: Filling Energy as a function of Hitch Height for bucket 12 in two spoils.

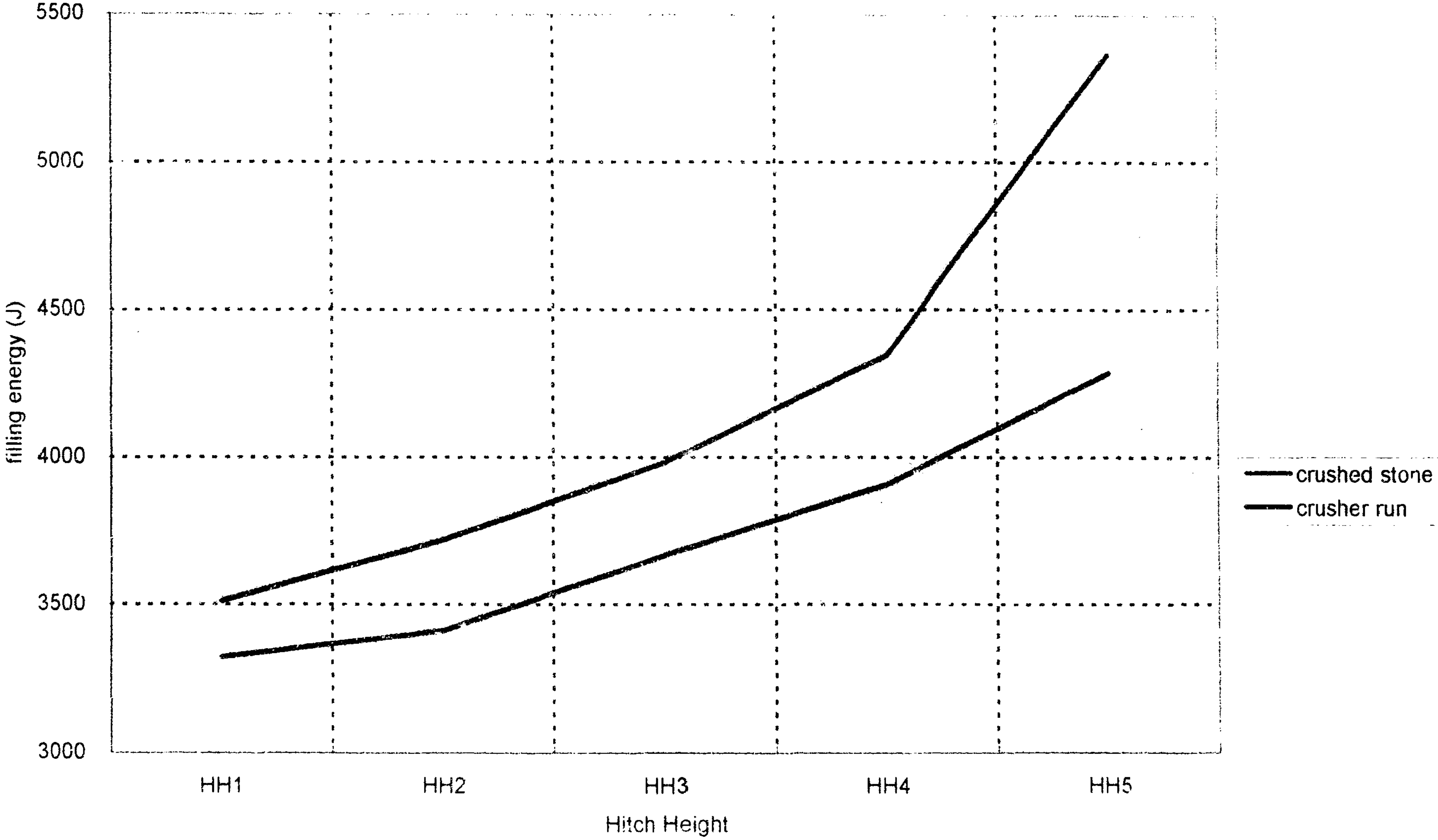

Figure C17: Filling Energy as a function of Hitch Height for bucket 11 in tw, spoils.

C17

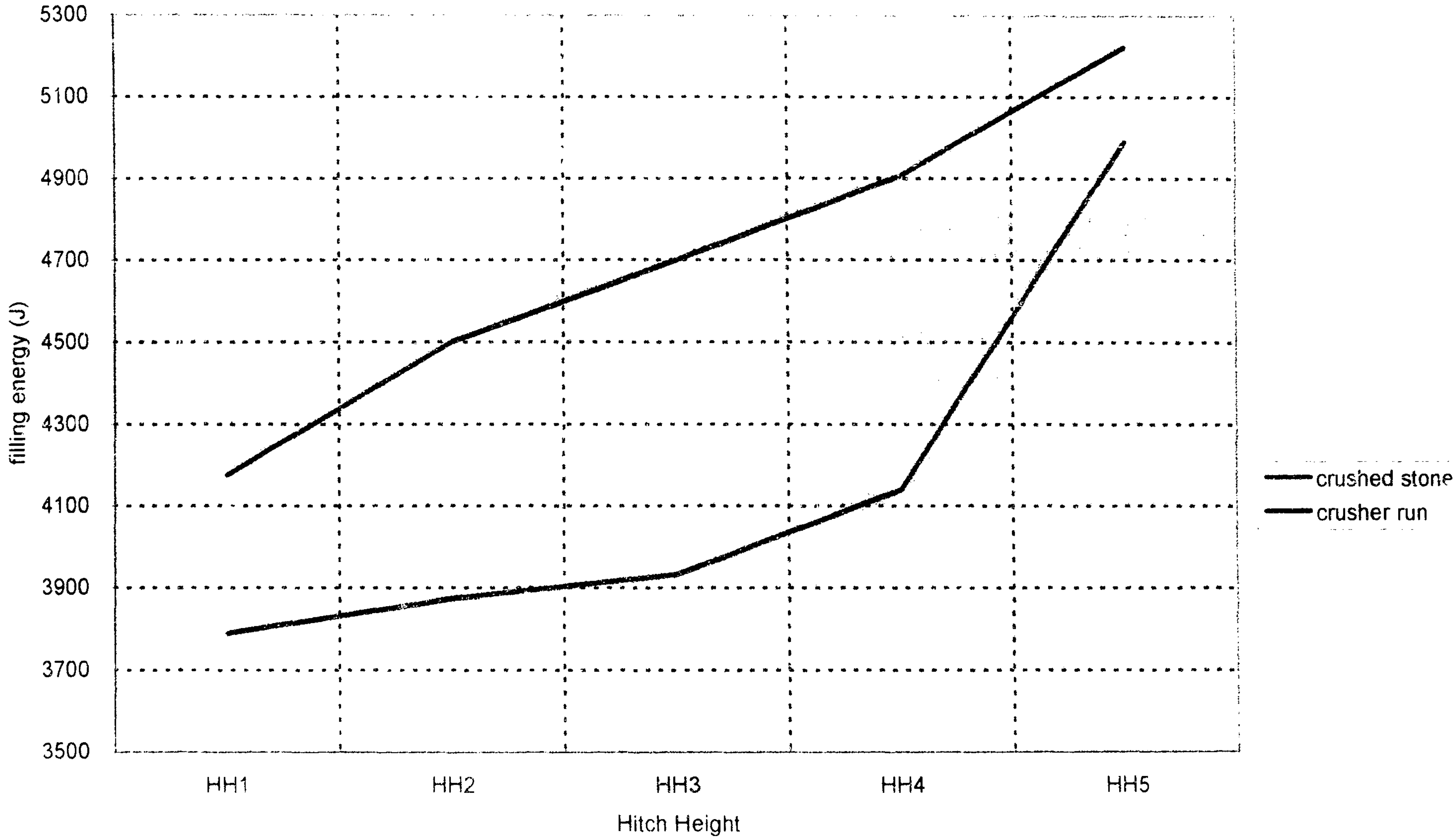

Figure C18: Filling Energy as a function of Hitch Height for bucket 10 in two spoils.

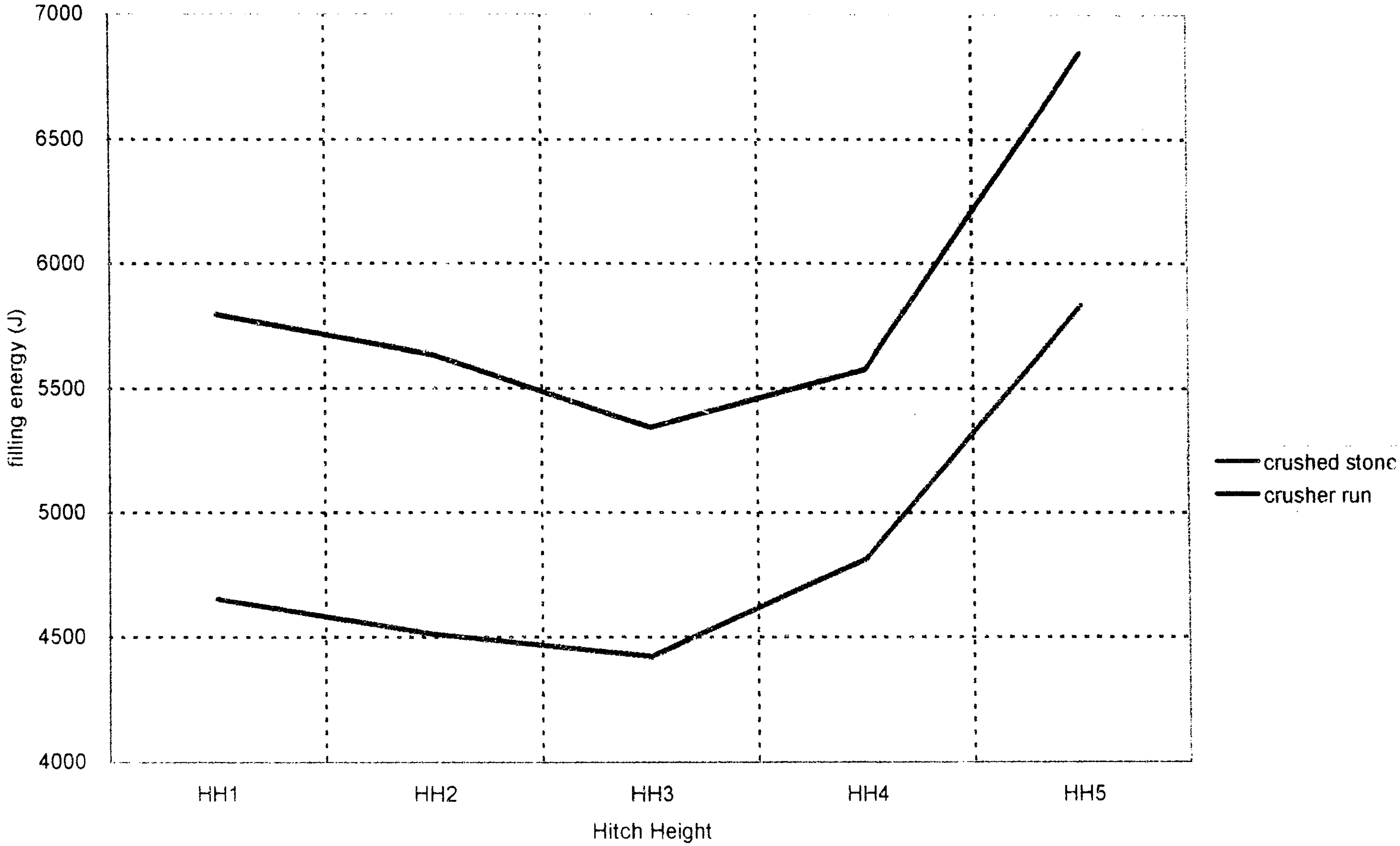

Figure C19: Filling Energy as a function of Hitch Height
for bucket 09 in two spoils.

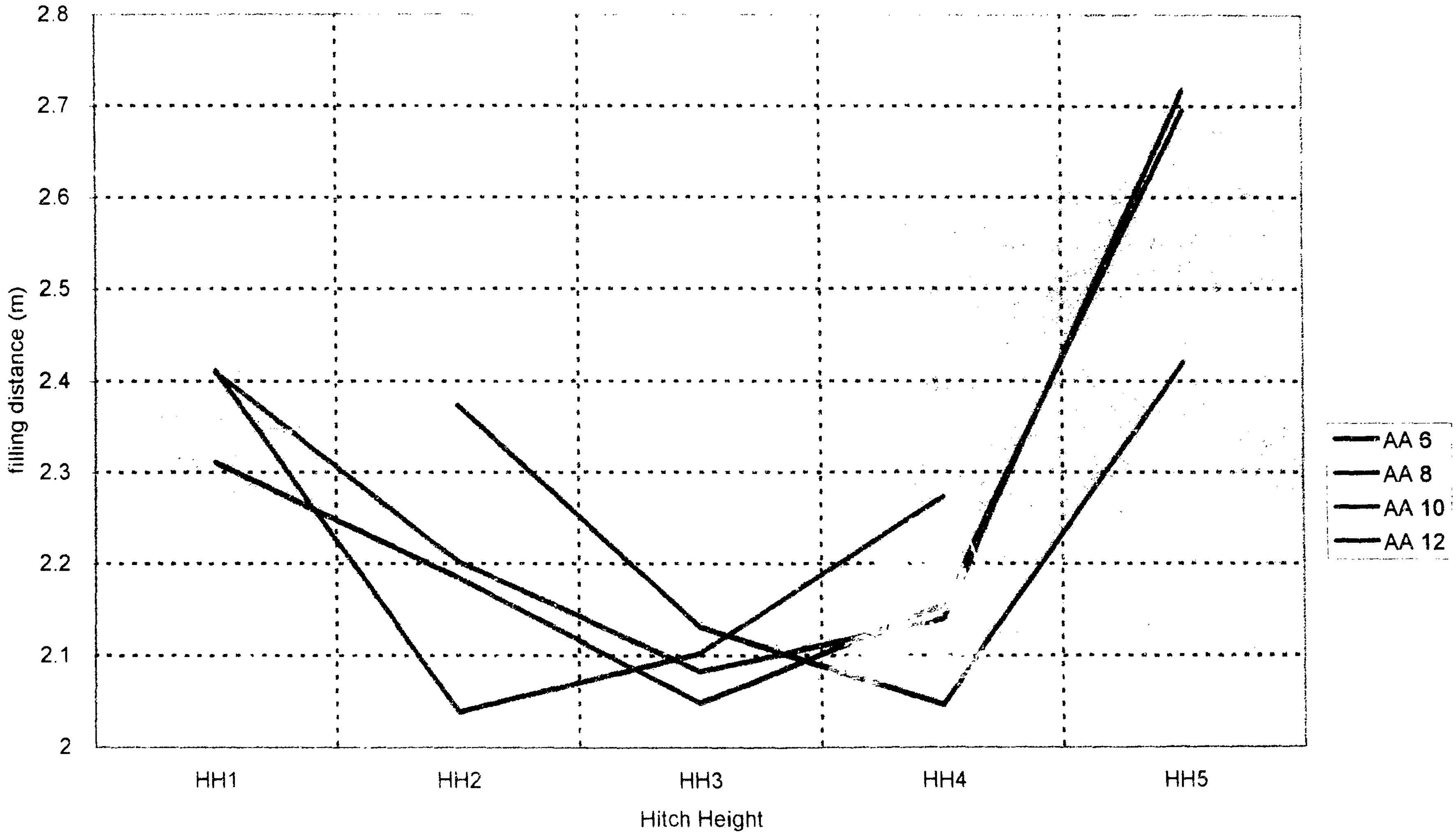

Figure C20: Filling Distance as a function of Hitch Height for four angles of attack for bucket 10 in crushed rock.

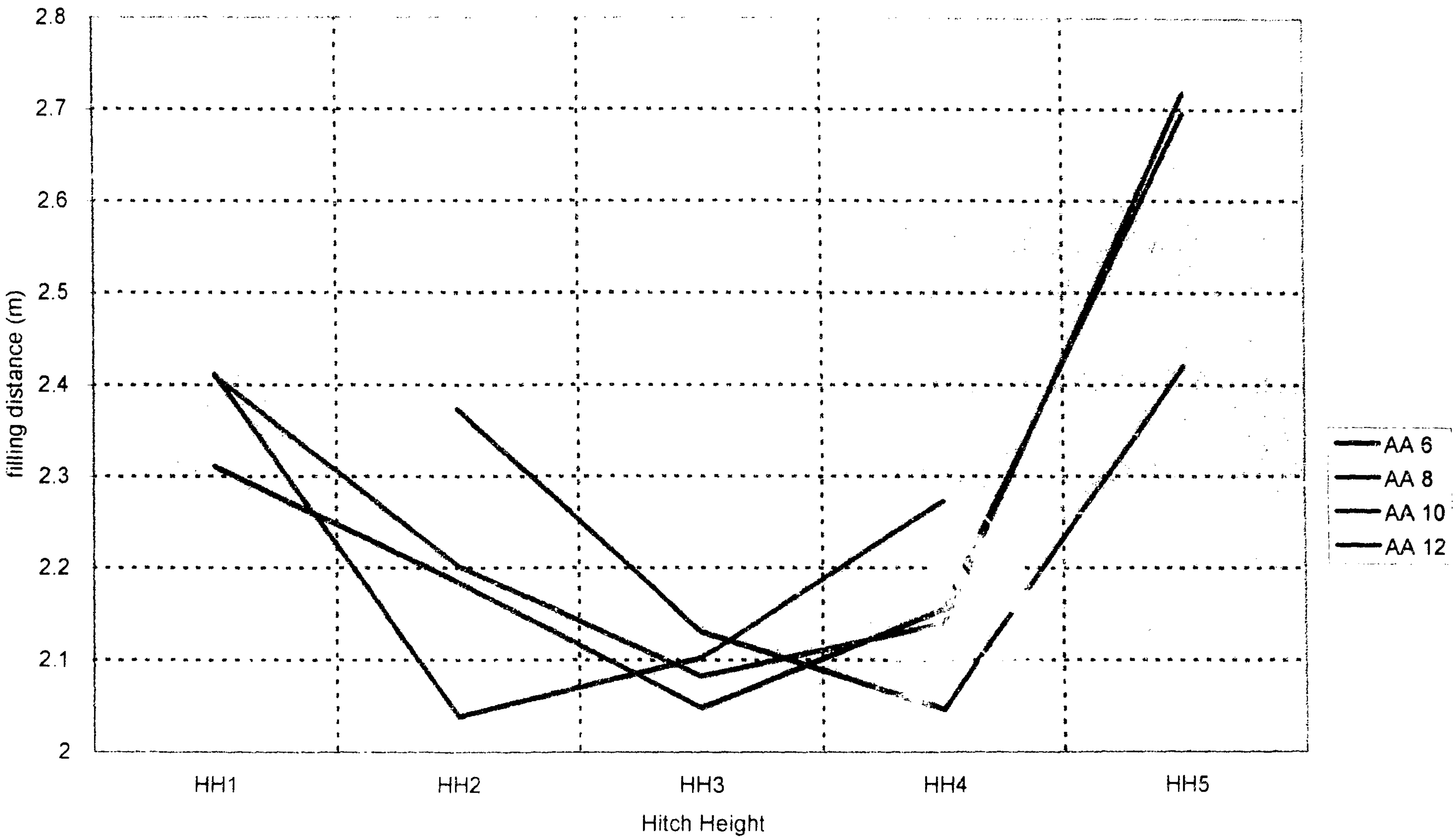

Figure C20: Filling Distance as a function of Hitch Height for four angles of attack for bucket 10 in crushed rock.

C20

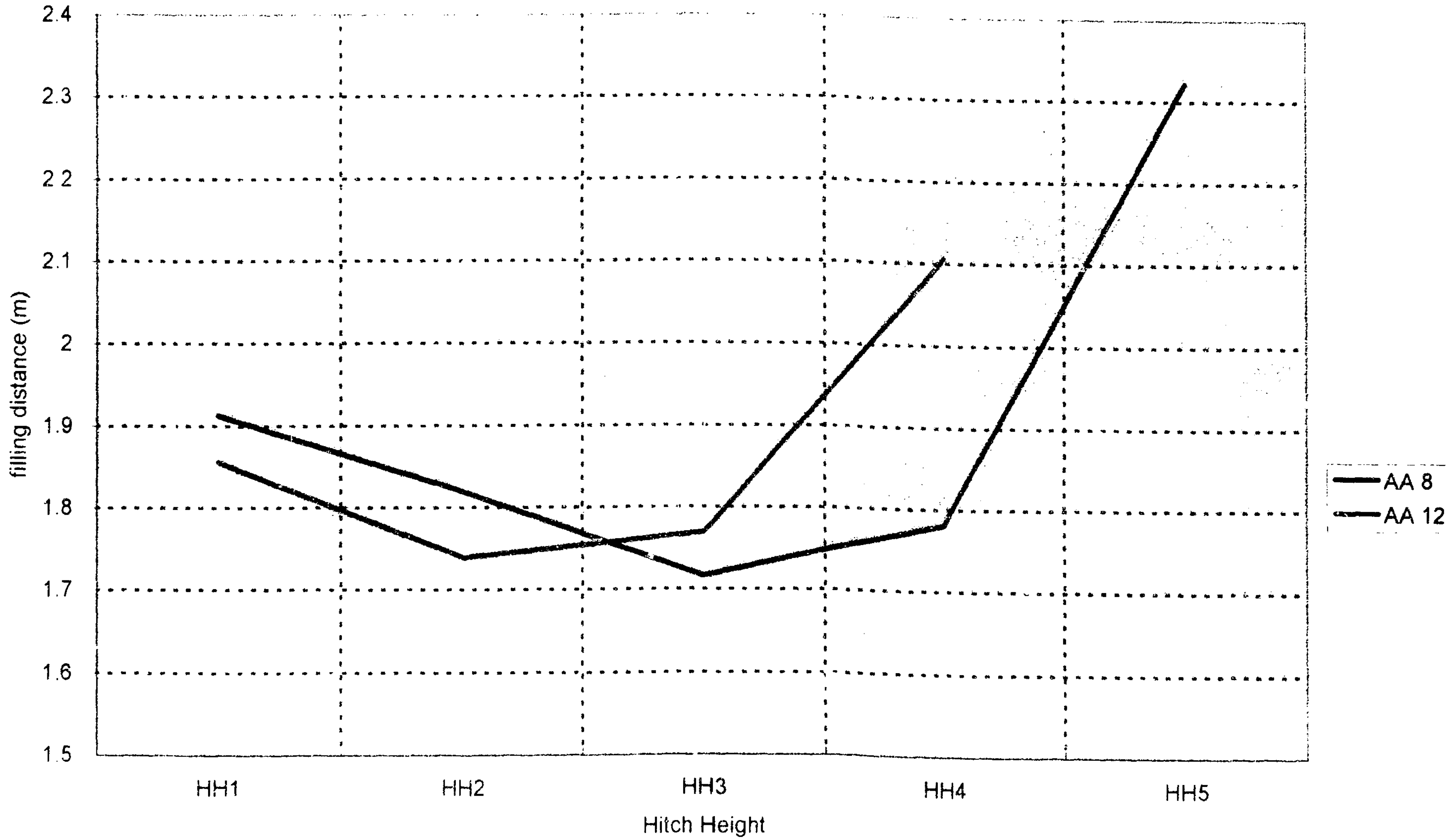

Figure C21: Filling Distance as a function of Hitch Height for two angles of attack for bucket 11 in crusher run.

C21

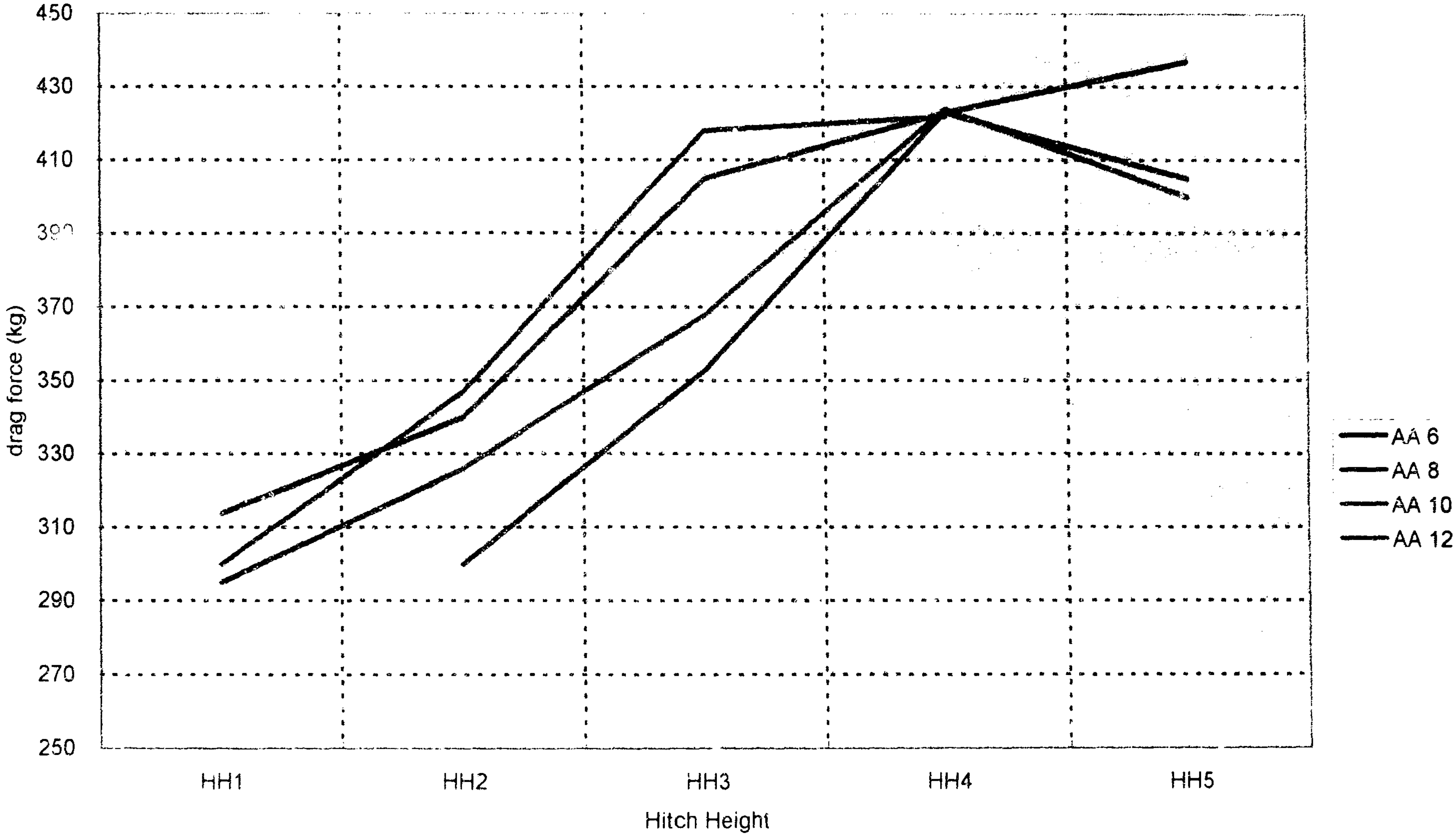

Figure C22: Max Drag Force as a function of Hitch Height for four angles of attack for bucket 10 in crushed rock.

C22

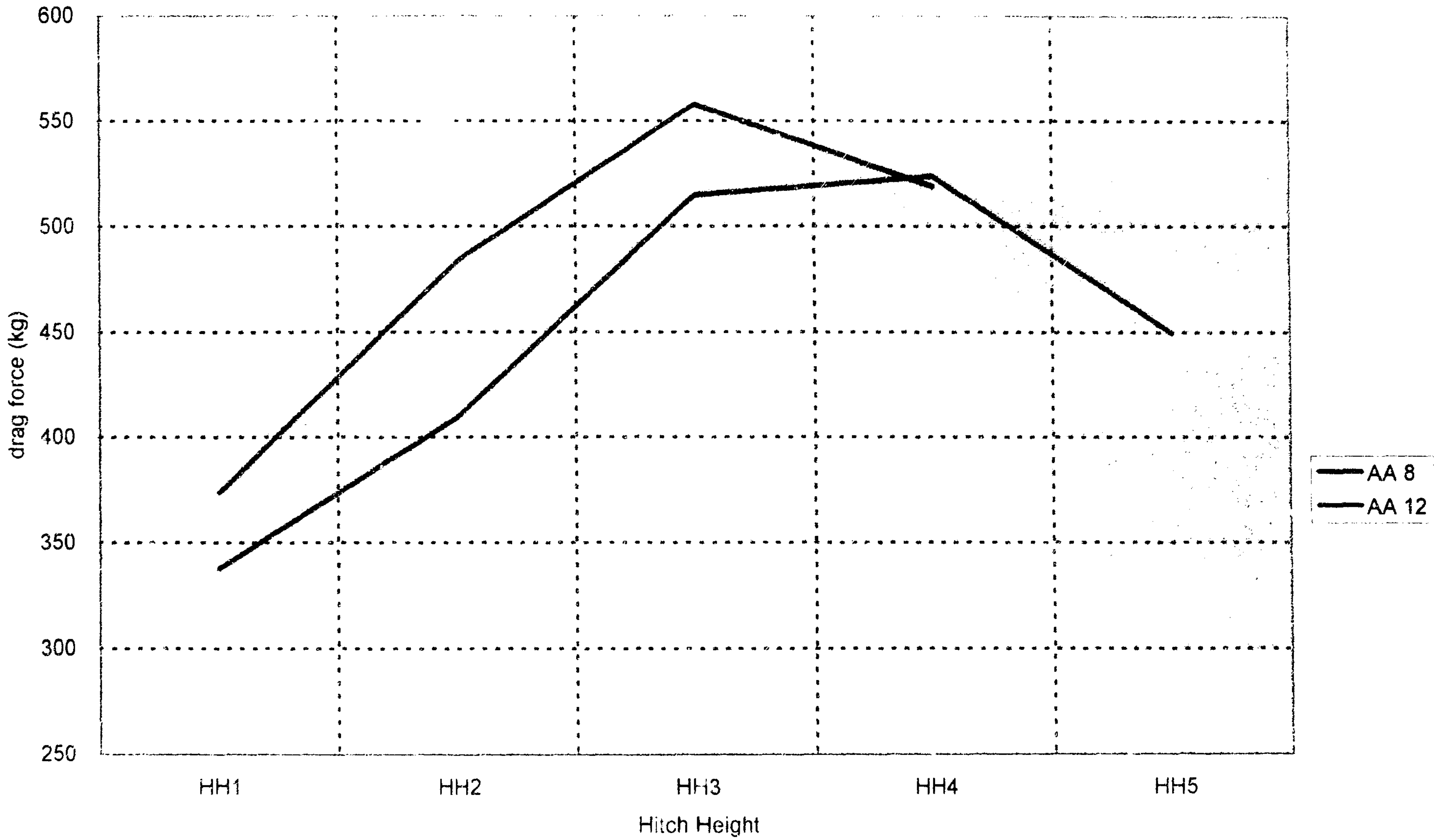

Figure C23: Max Drag Force as a function of Hitch Height for two angles of attack for bucket 11 in crusher run.

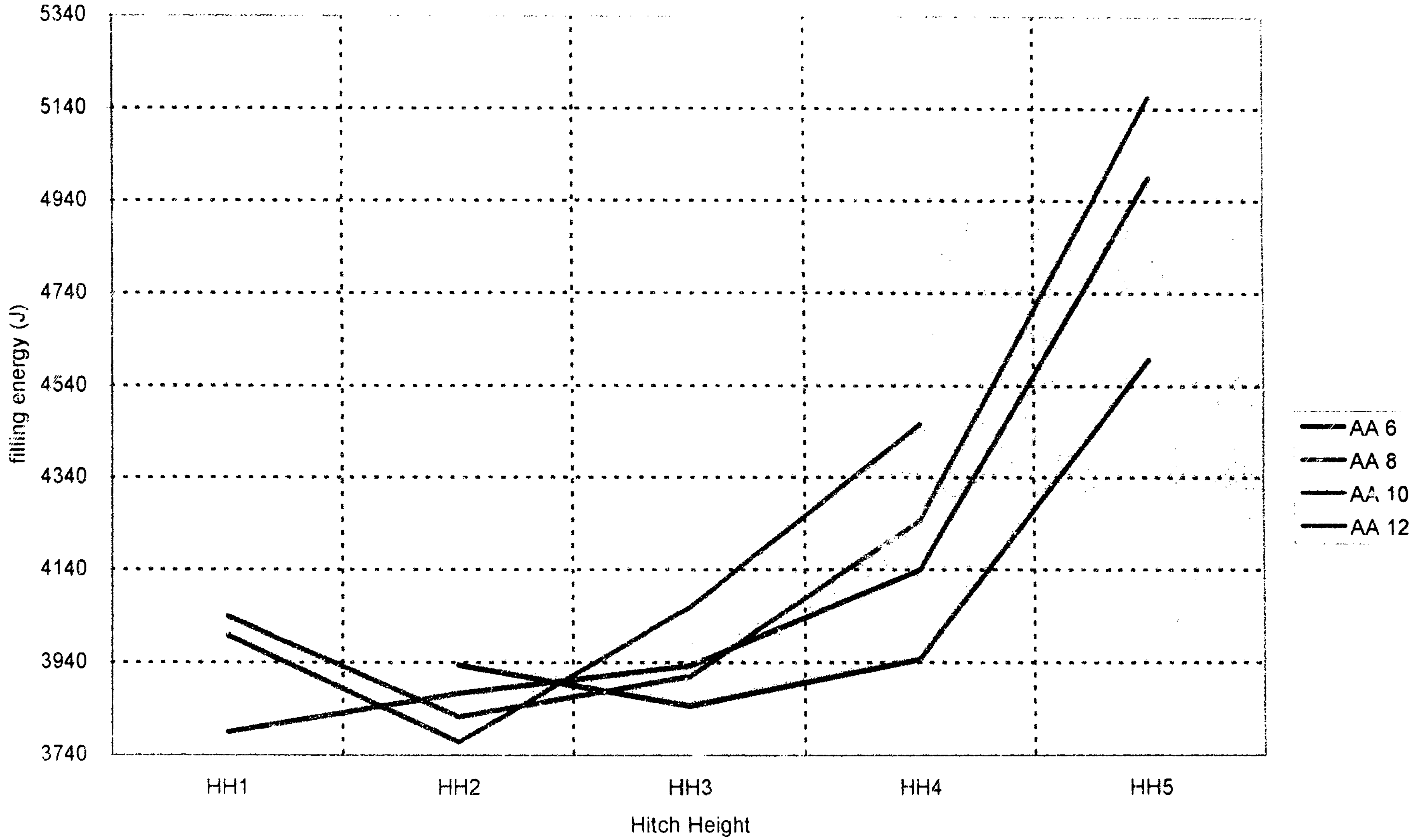

Figure C24: Filling Energy as a function of Hitch Height for four angles of attack for bucket 10 in crushed rock.

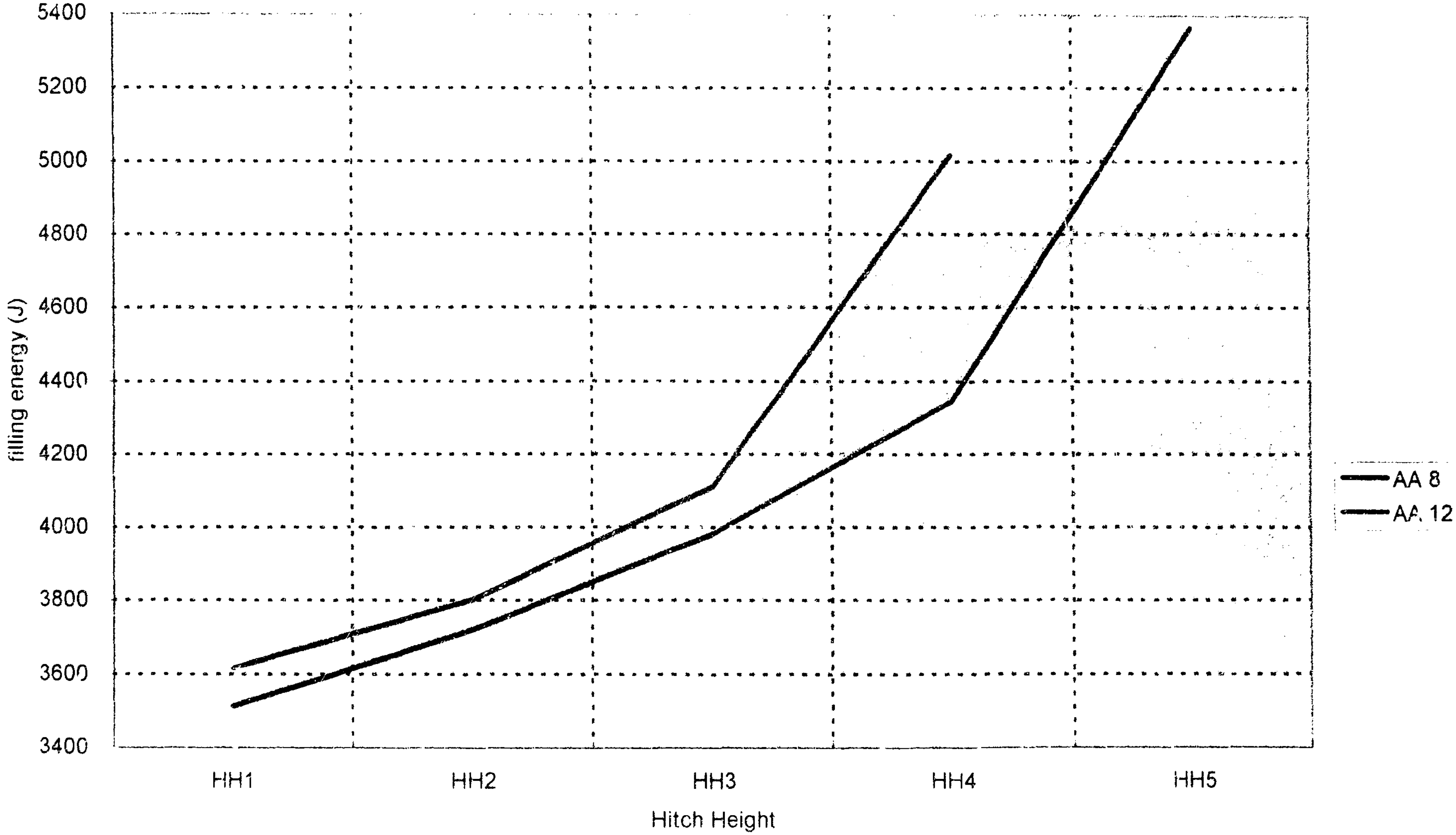

Figure C25: Filling Energy as a function of Hitch Height for two angles of attack for bucket 11 in crusher run.

C25

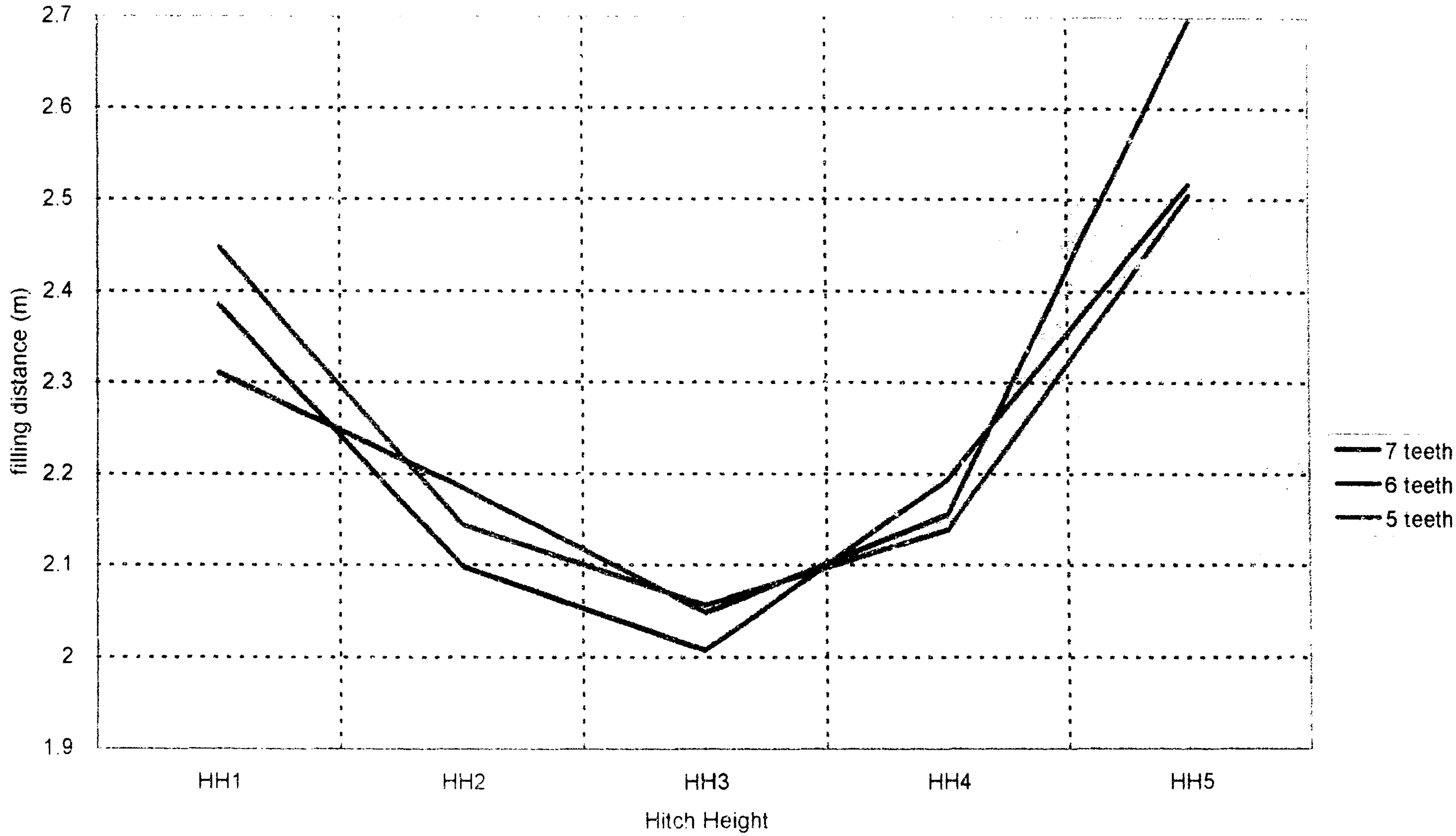

Figure C26: Filling Distance as a function of Hitch Height for bucket 10 in crushed rock with a 8 deg angle of attack and different numbers of teeth.

C26

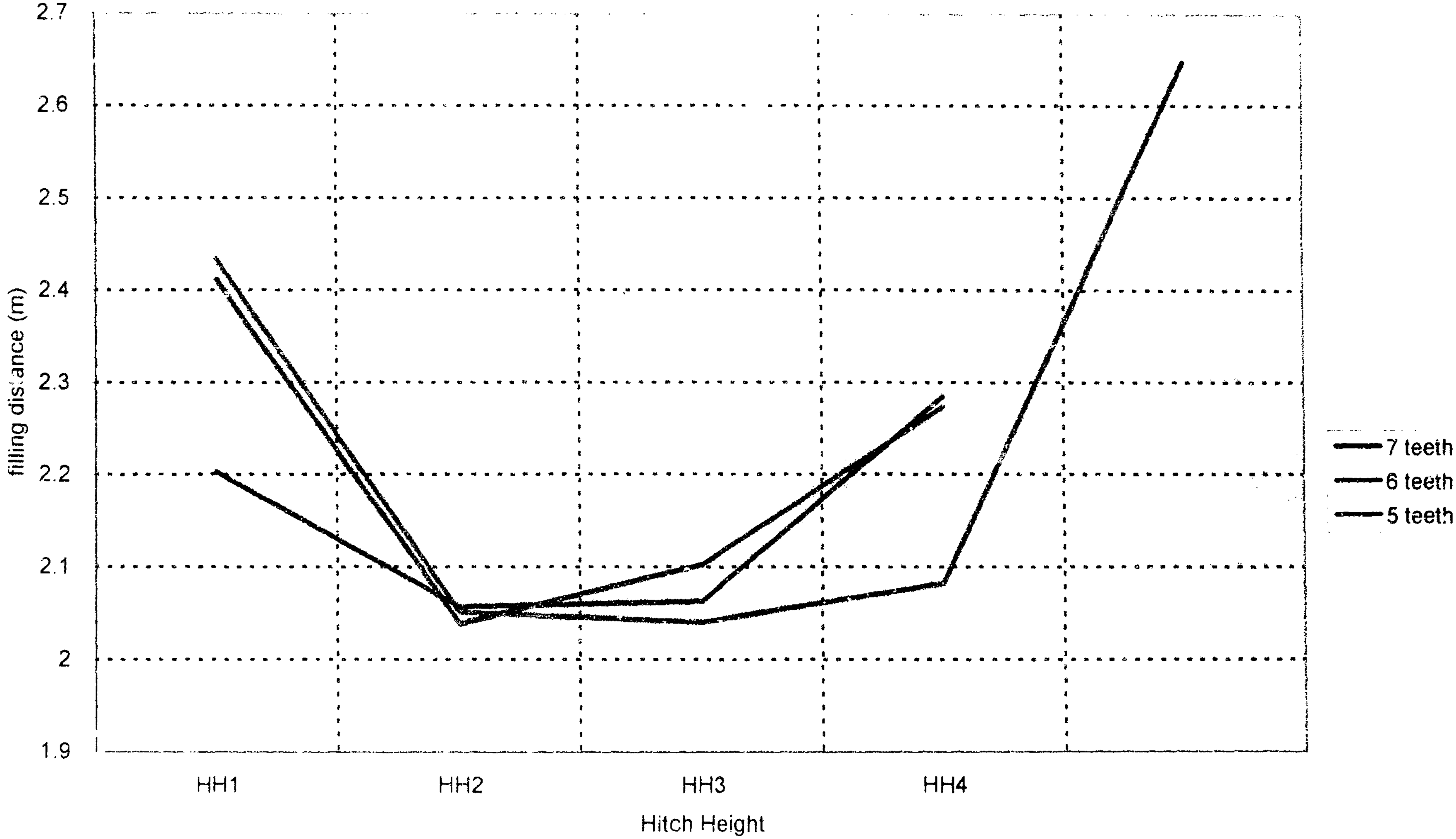

Figure C27: Filling Distance as a function of Hitch Height for bucket 10 in crushed rock with a 12 deg angle of attack and different numbers of teeth.

Figure C28: Max Drag Force as a function of Hitch Height for bucket 10 in crushed rock with a 8 deg angle of attack and different numbers of teeth.

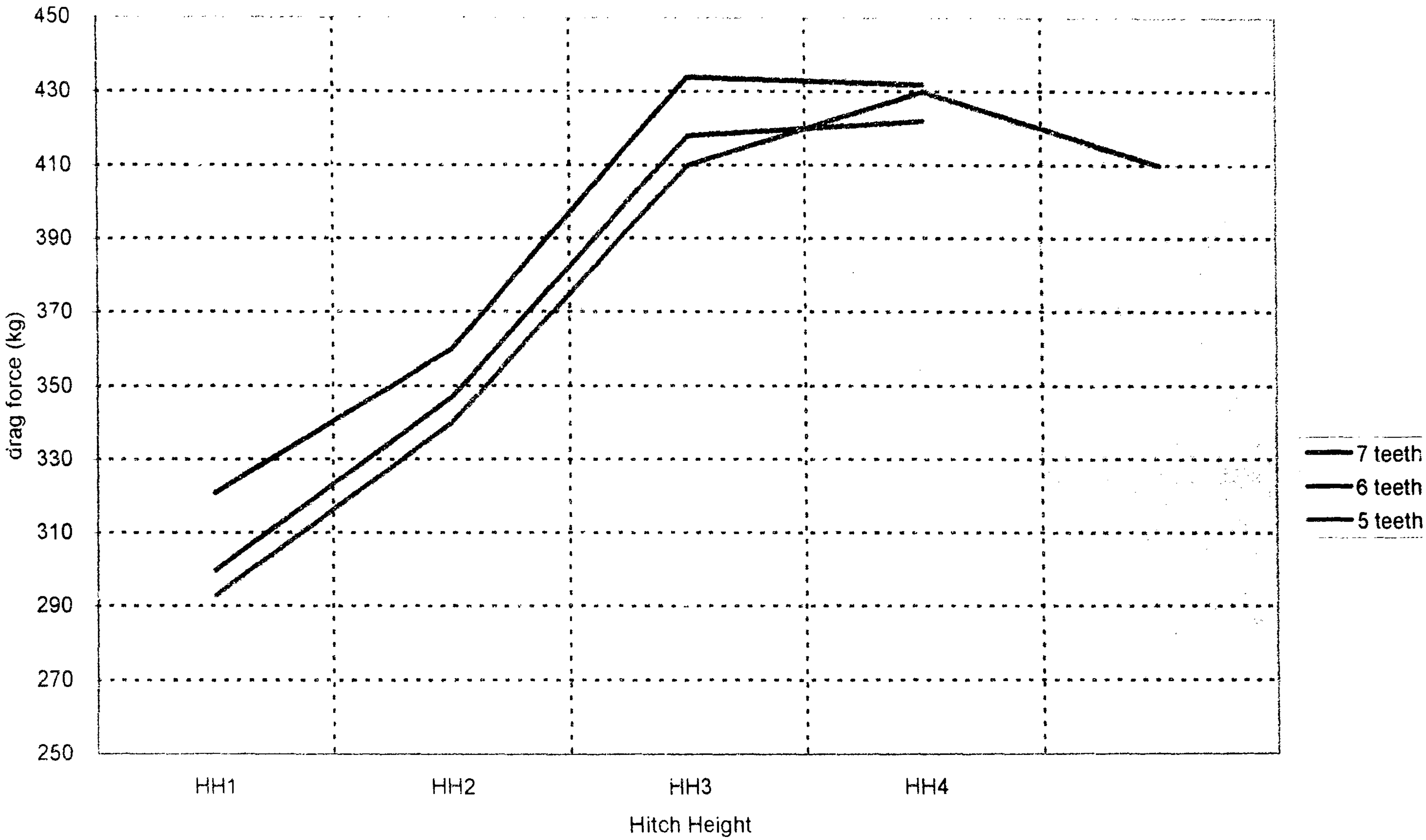

Figure C29: Max Drag Force as a function of Hitch Height for bucket 10 in crushed rock with a 12 deg angle of attack and different numbers of teeth.

C29

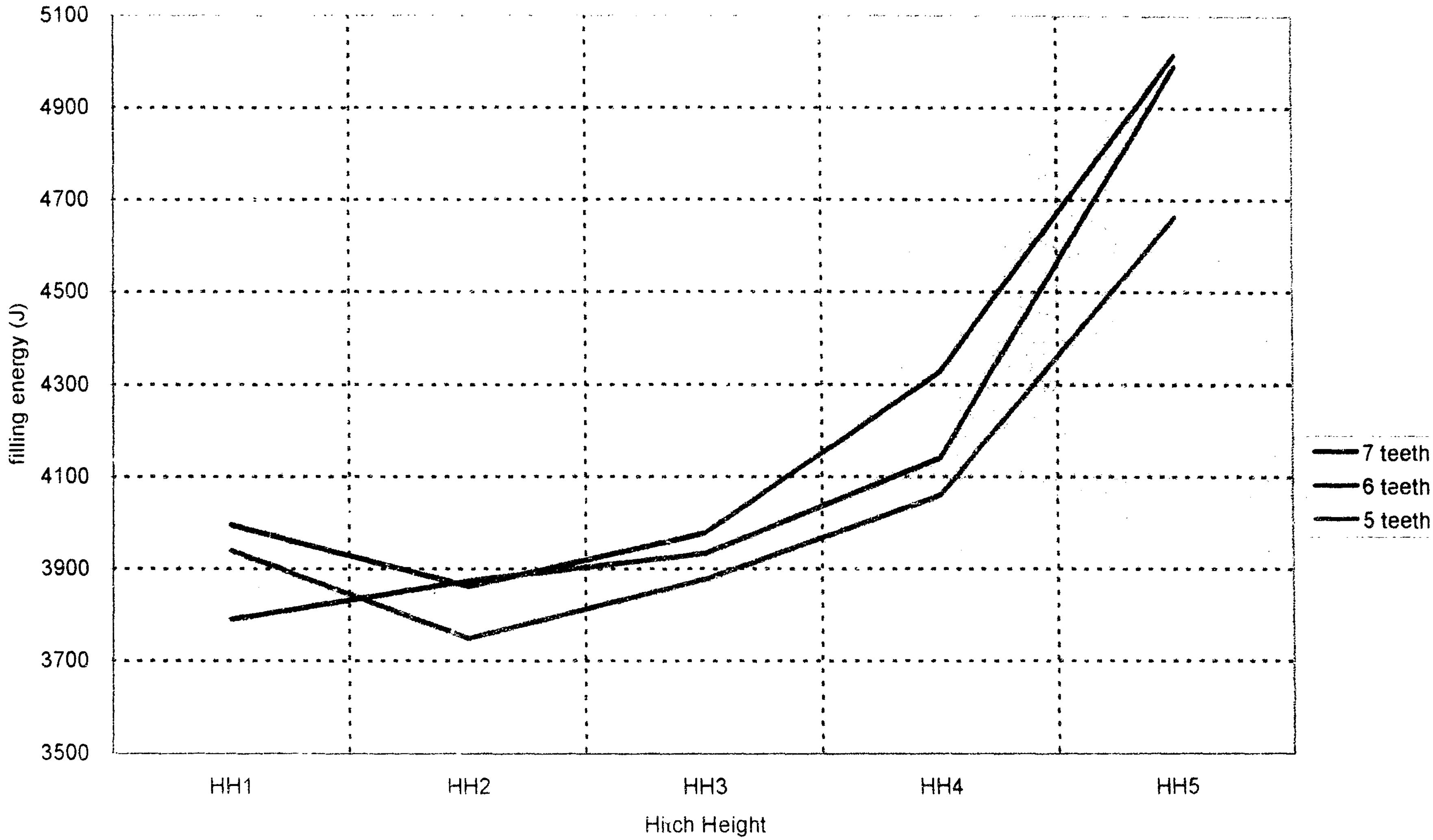

Figure C30: Filling Energy as a function of Hitch Height for bucket 10 in crushed rock with a 8 deg angle of attack and different numbers of teeth.

C31

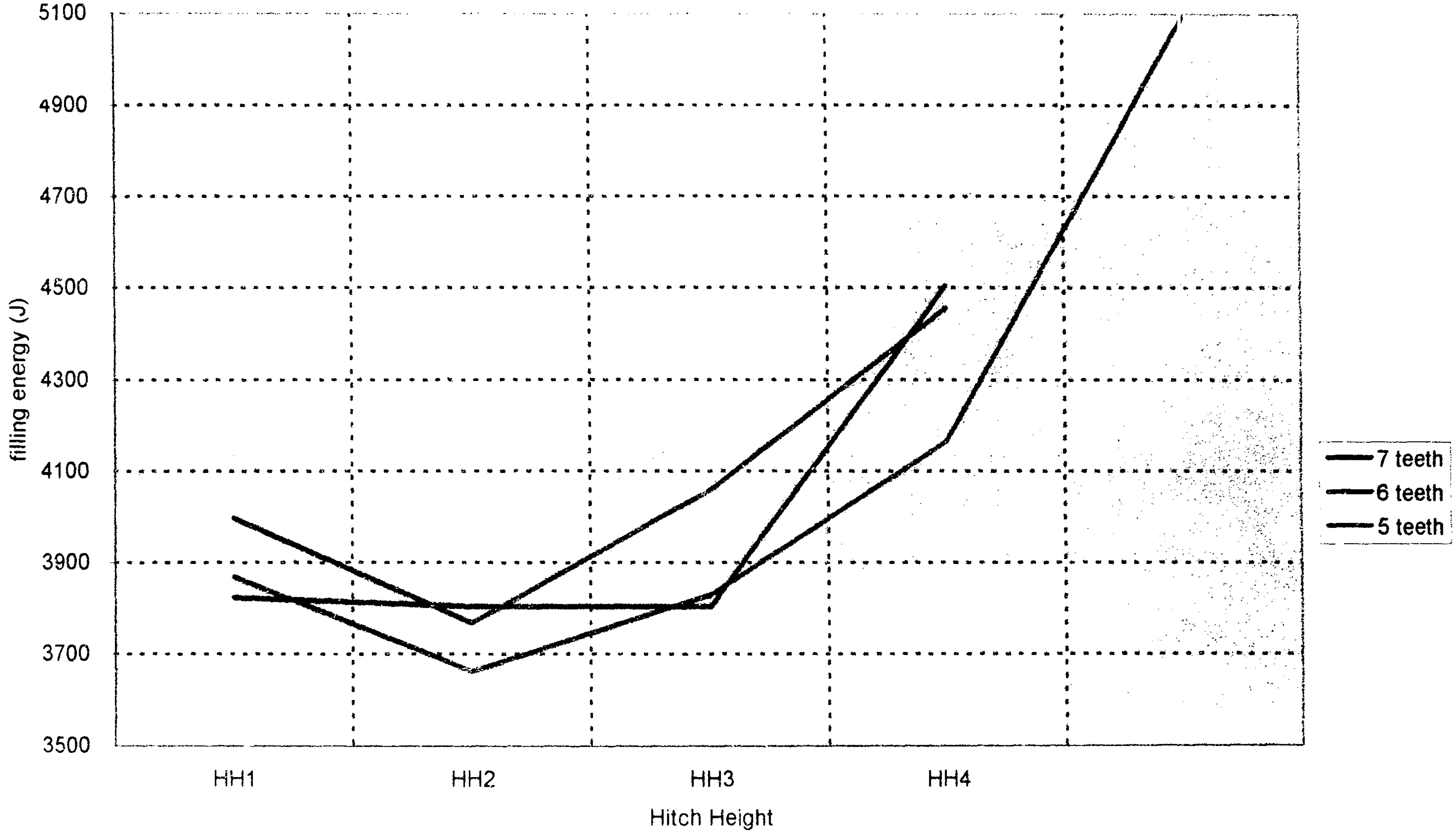

Figure C31: Filling Energy as a function of Hitch Height for bucket 10 in crushed rock with a 12 deg angle of attack and different numbers of teeth.

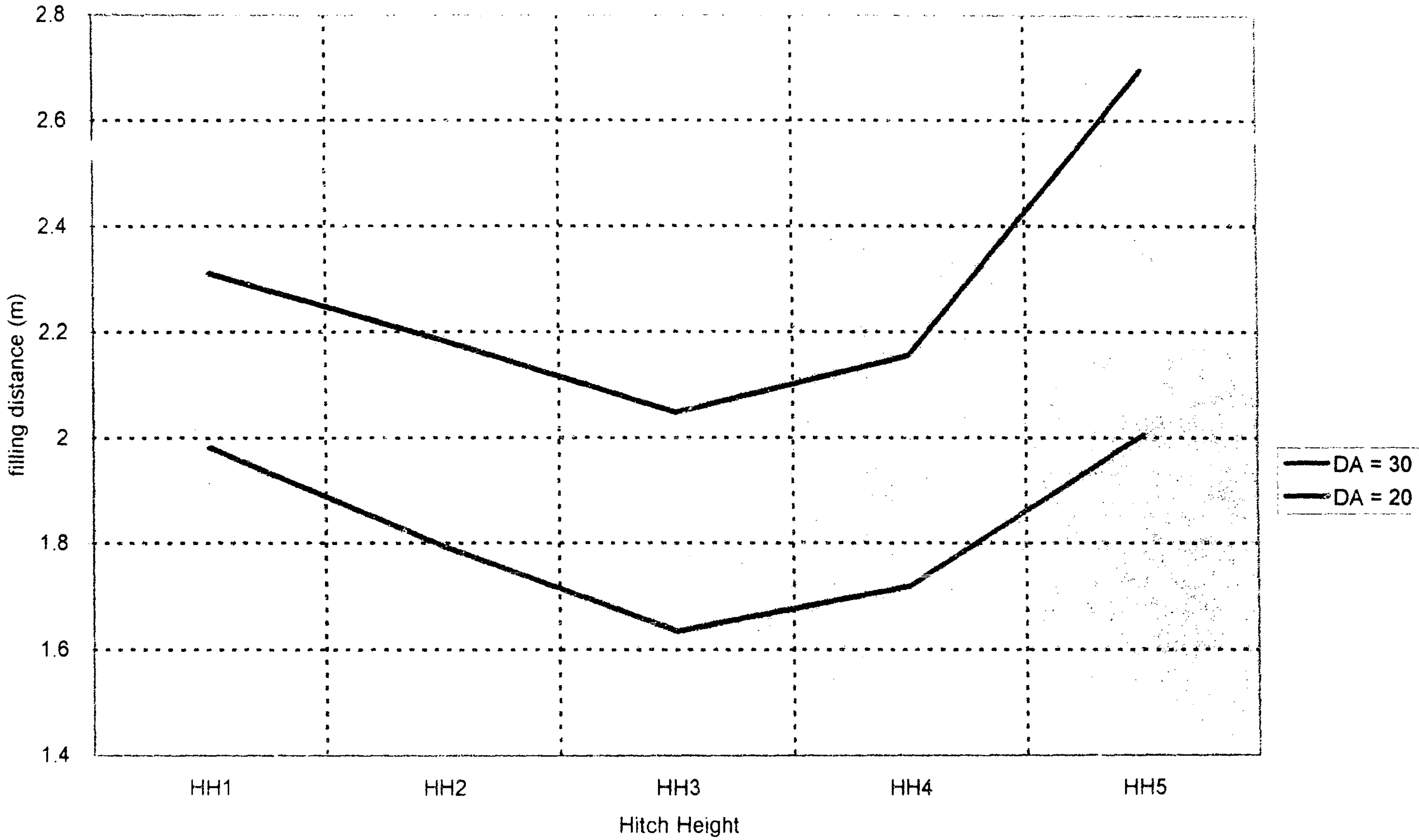

Figure C32: Filling Distance as a function of Hitch Height for bucket 10 at two drag angles in crushed stone.

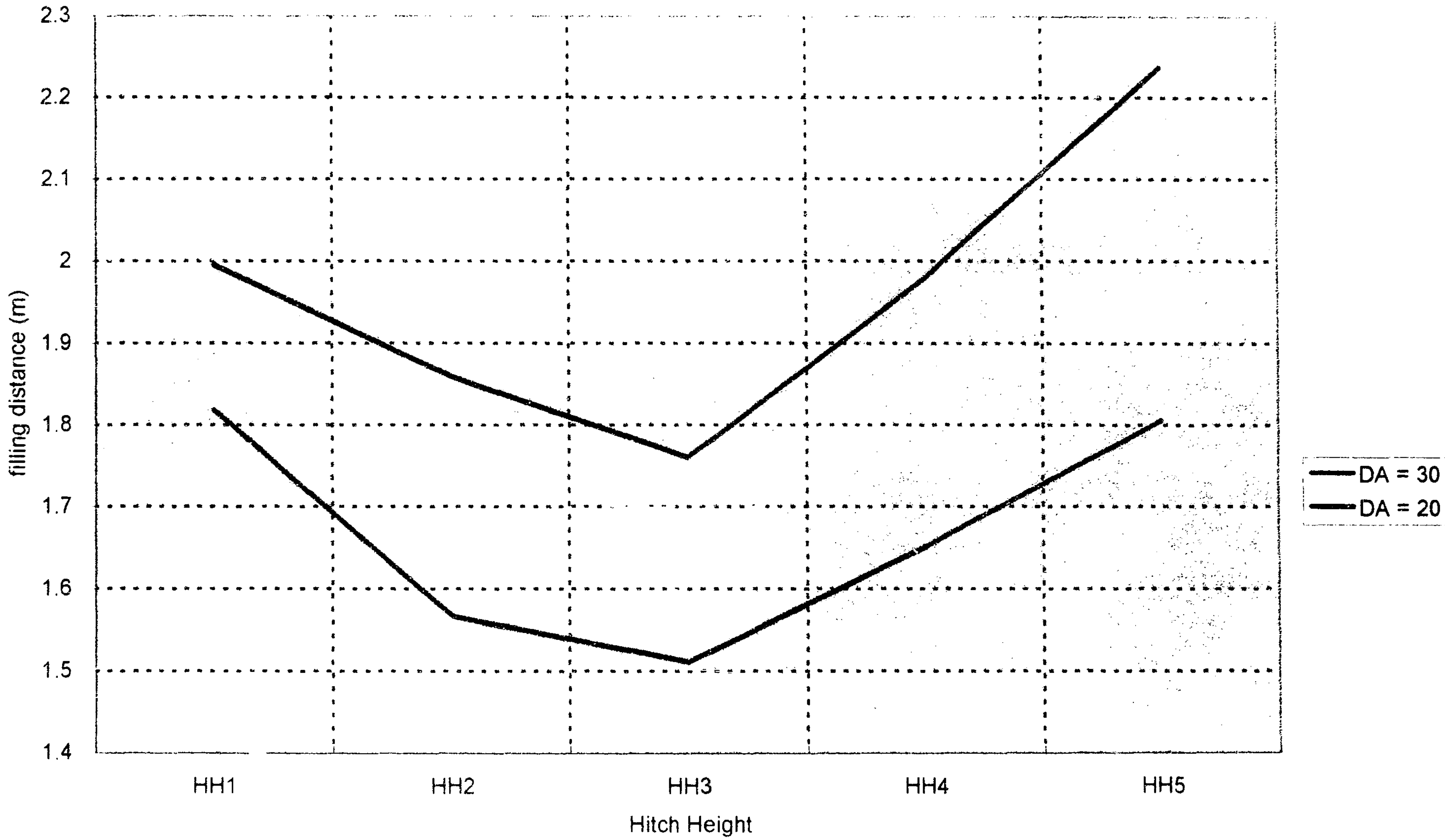

Figure C33: Filling Distance as a function of Hitch Height for bucket 12 at two drag angles in crushed stone.

C33

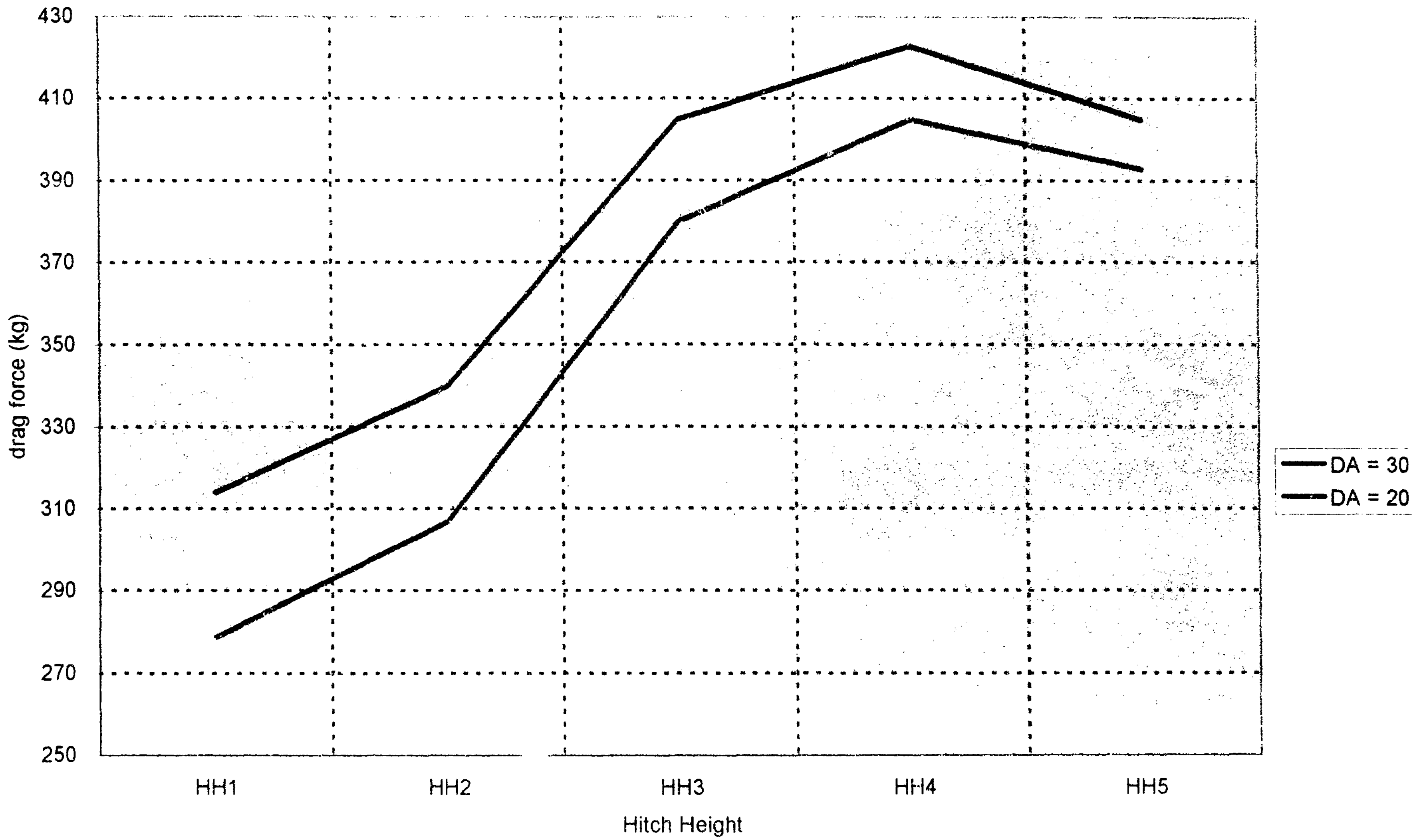

Figure C34: Max Drag Force as a function of Hitch Height for bucket 10 at two drag angles in crushed stone.

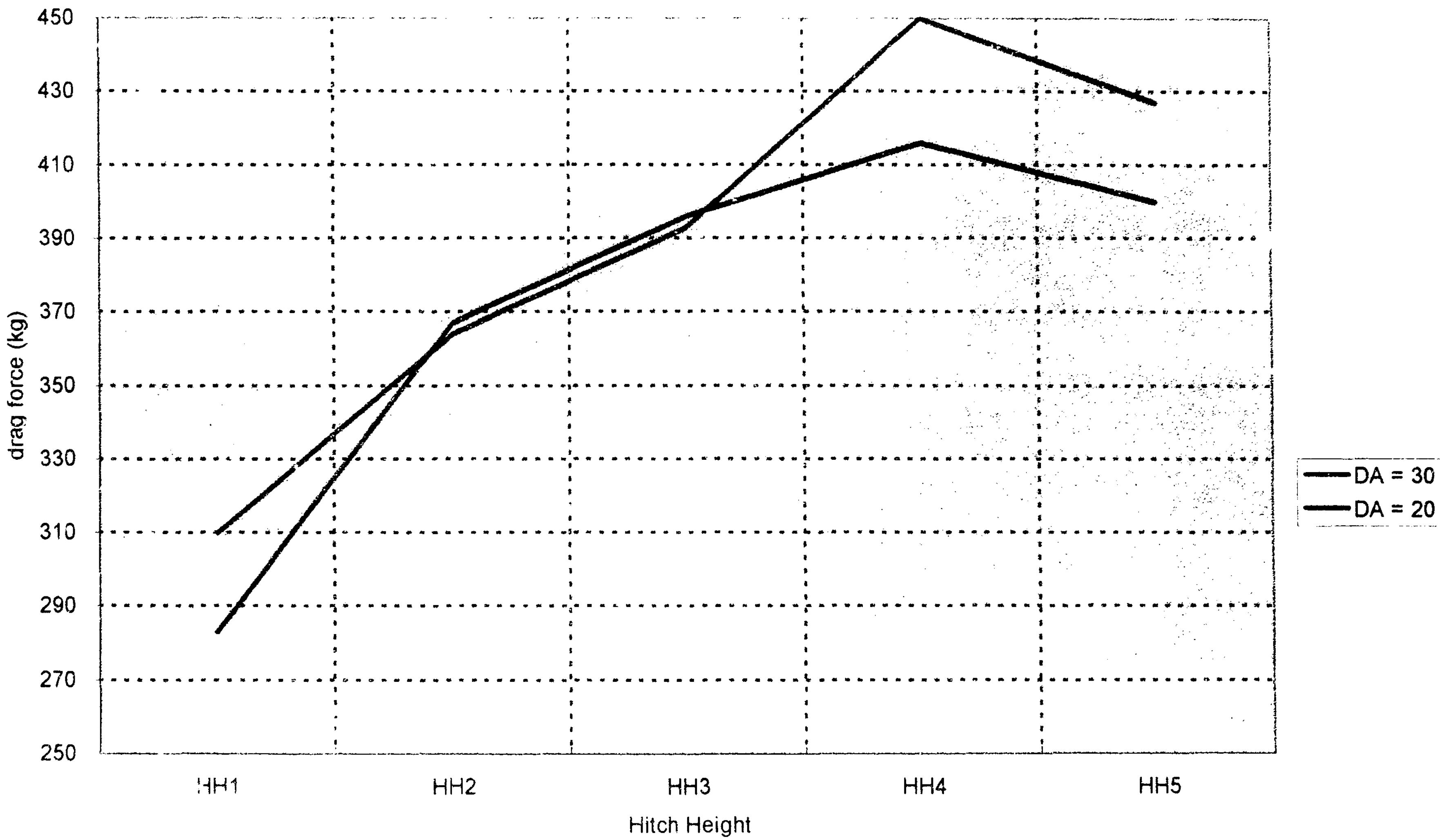

Figure C35: Max Drag Force as a function of Hitch Height for bucket 12 at two drag angles in crushed stone.

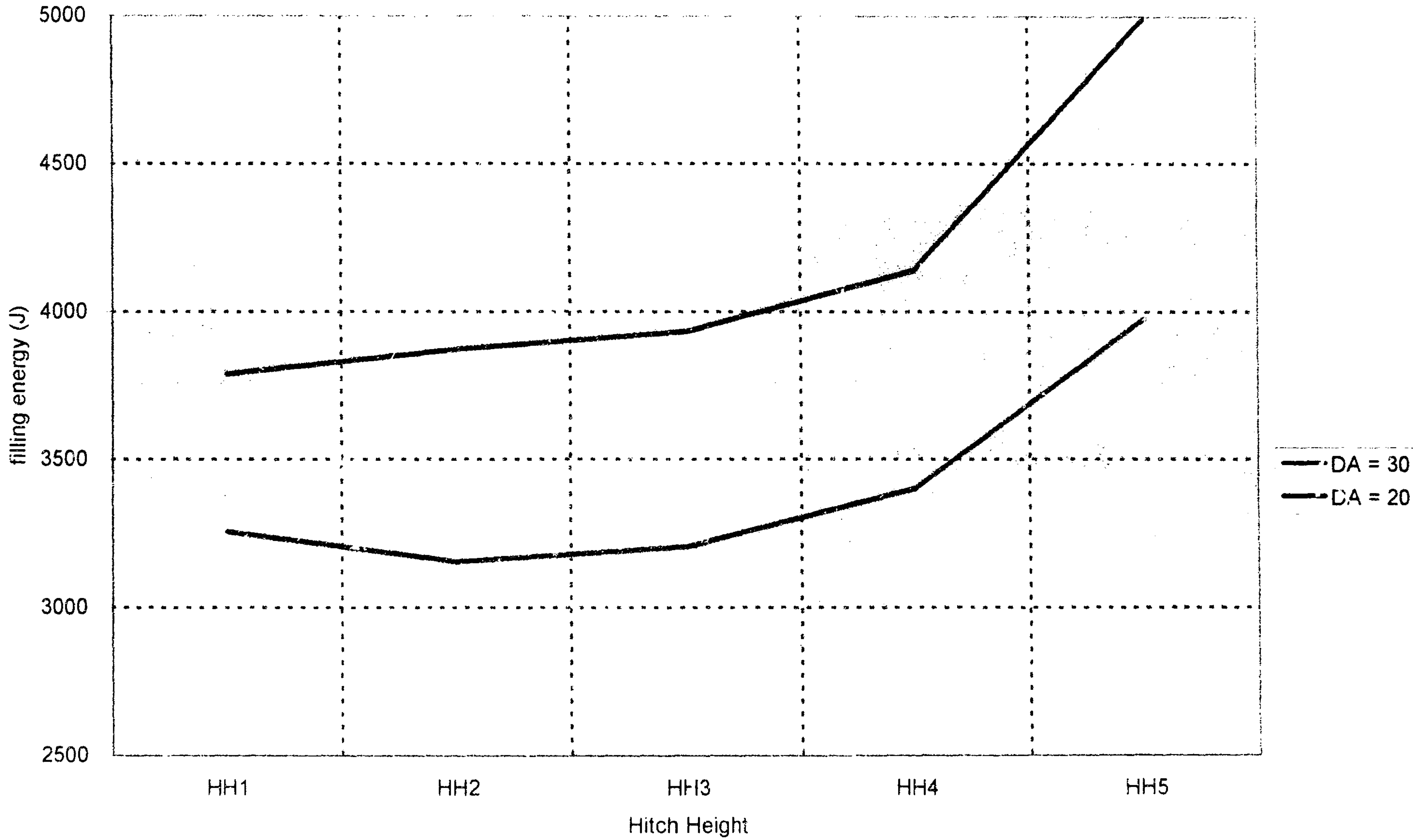

Figure C36: Filling Energy as a function of Hitch Height for bucket 10 at two drag angles in crushed stone.

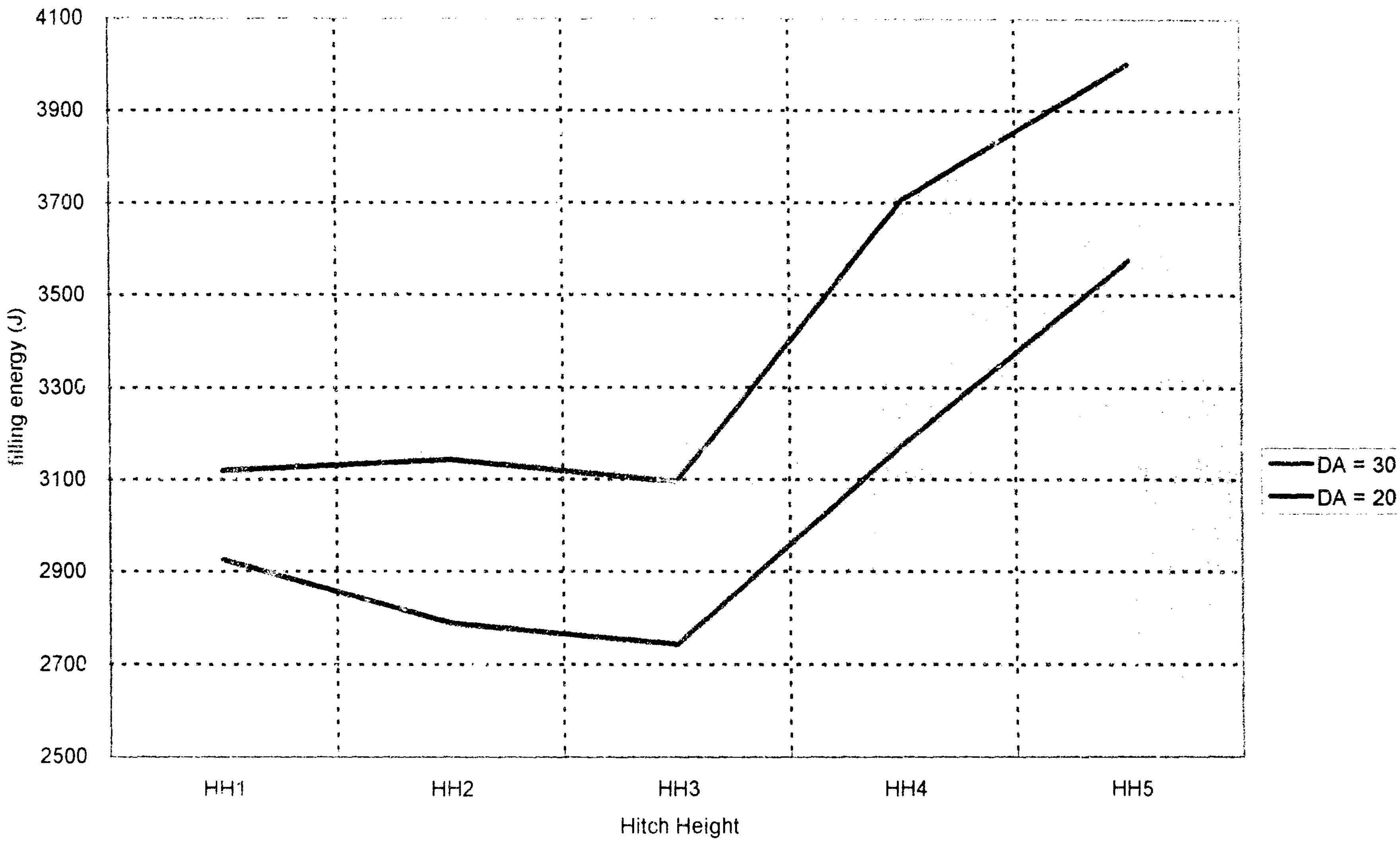

Figure 37: Filling Energy as a function of Hitch Height for bucket 12 at two drag angles in crushed stone.

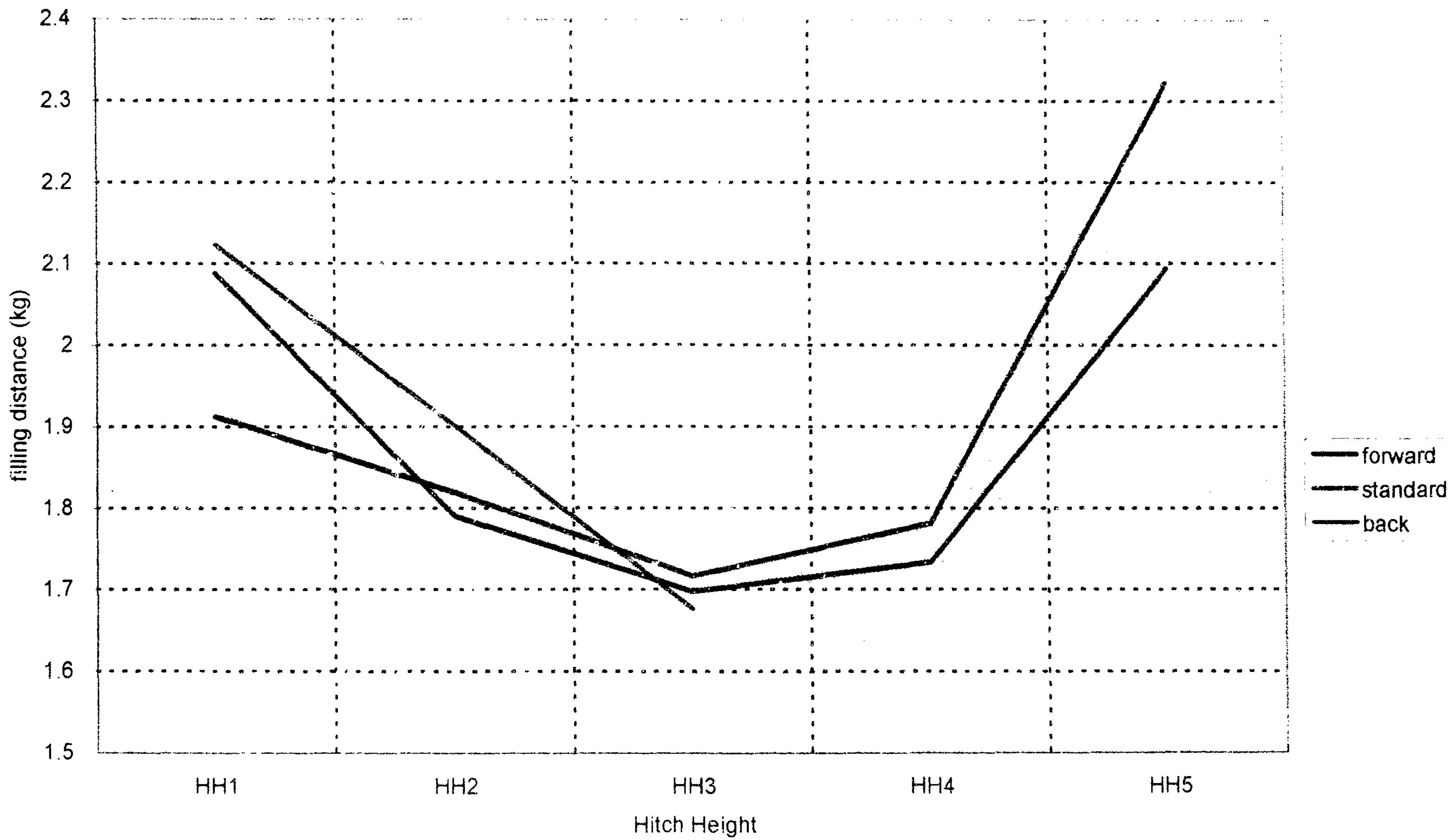

Figure C38: Filling Distance as a function of Hitch Height for a shift of the hitch in a horizontal direction

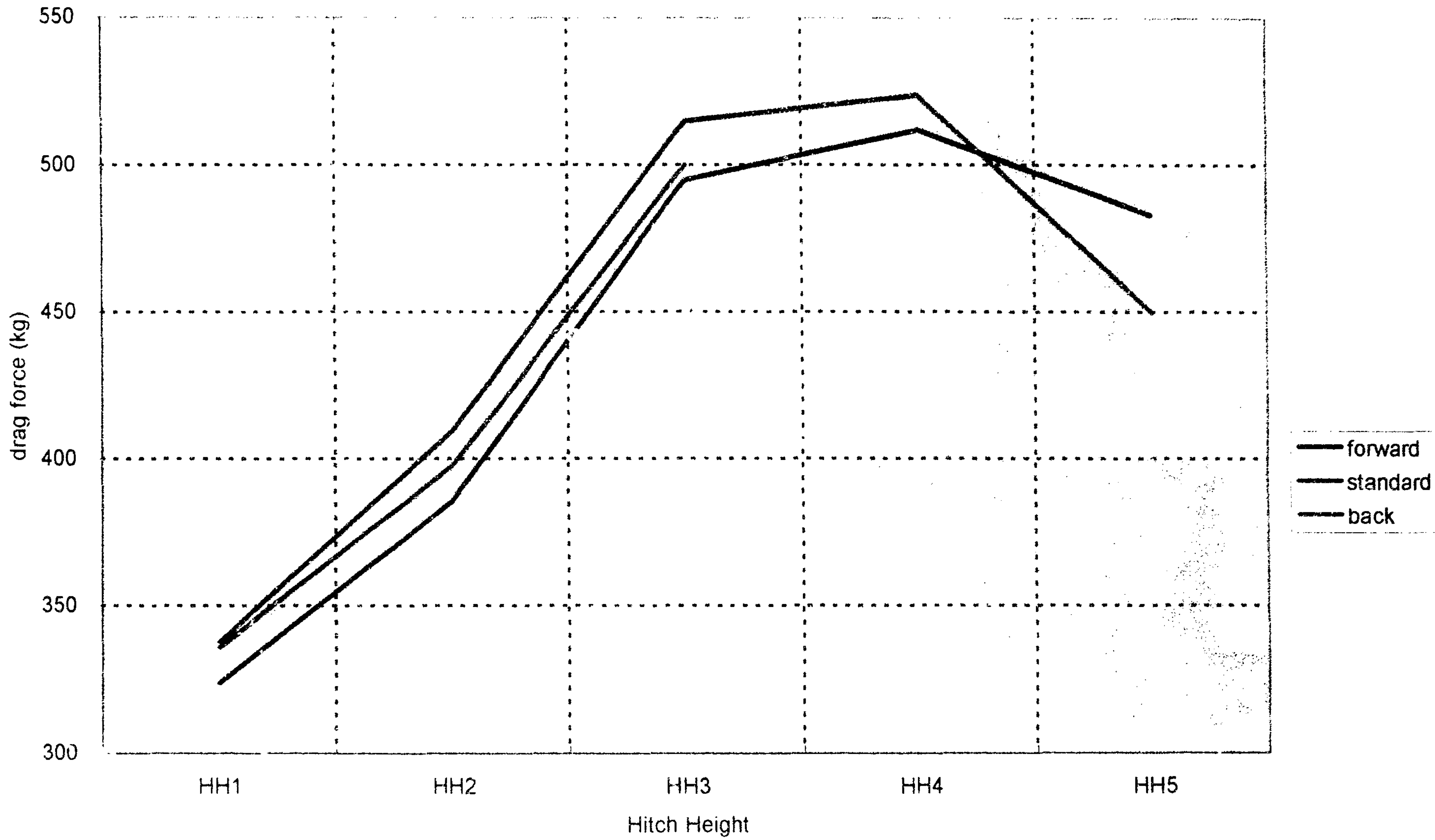

Figure C39: Max Drag Force as a function of Hitch Height
for a shift of the hitch in a horizontal direction

C39

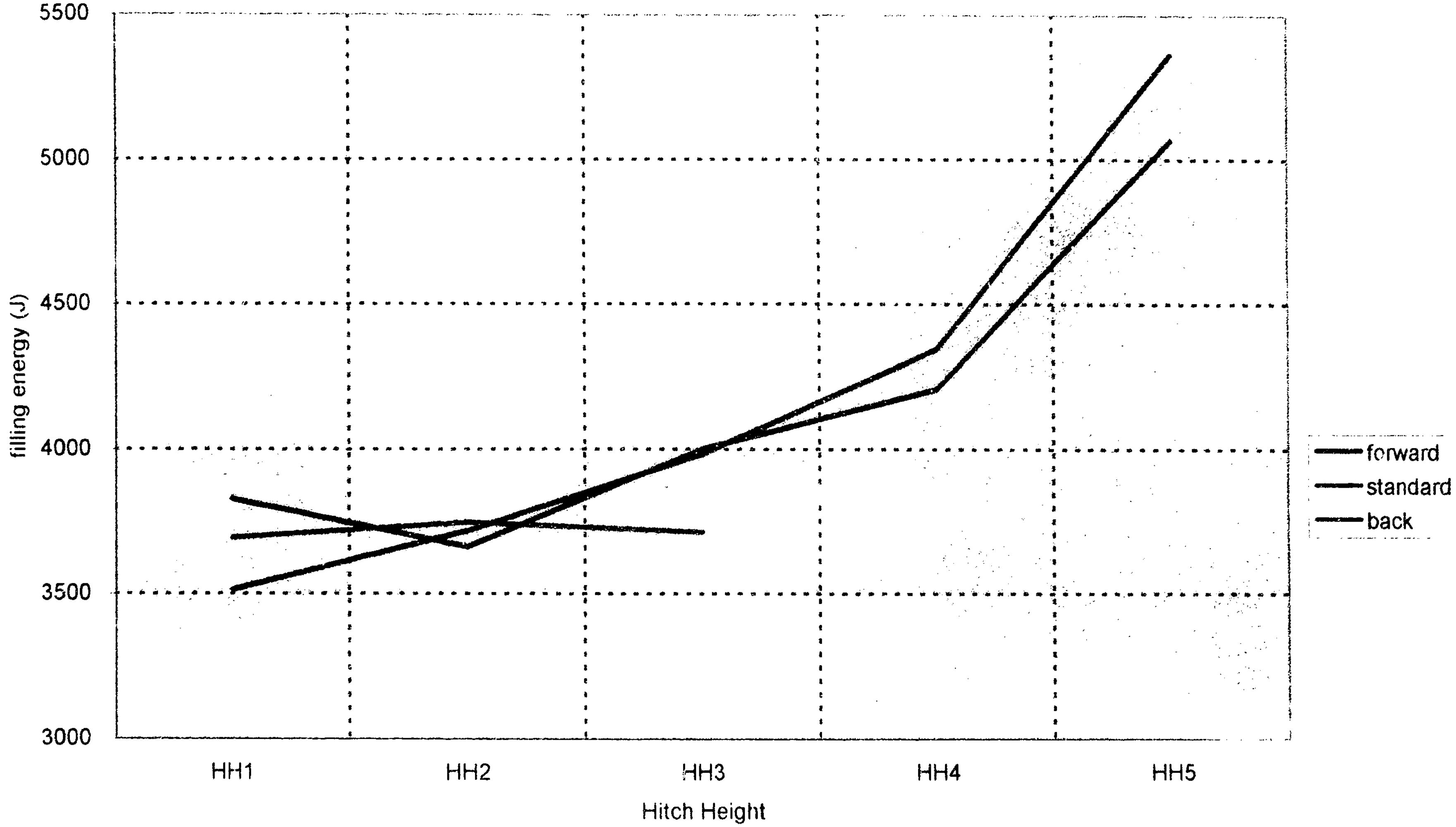

Figure C40: Filling Energy as a function of Hitch Height for a shift of the hitch in a horizontal direction

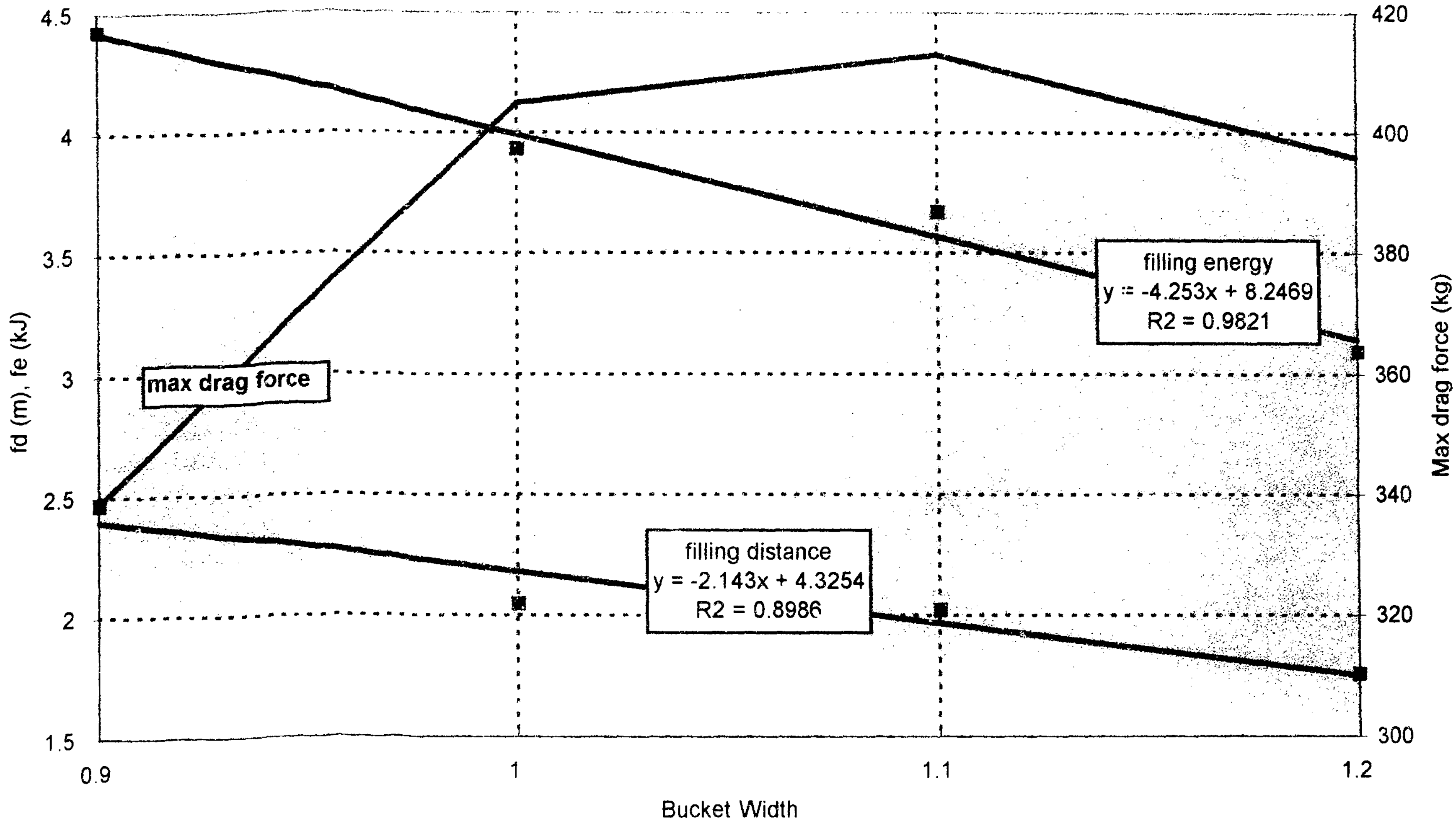

Figure C41: Relationships for performance parameters as function of bucket width on hitch height 3 in crushed rock.

C42

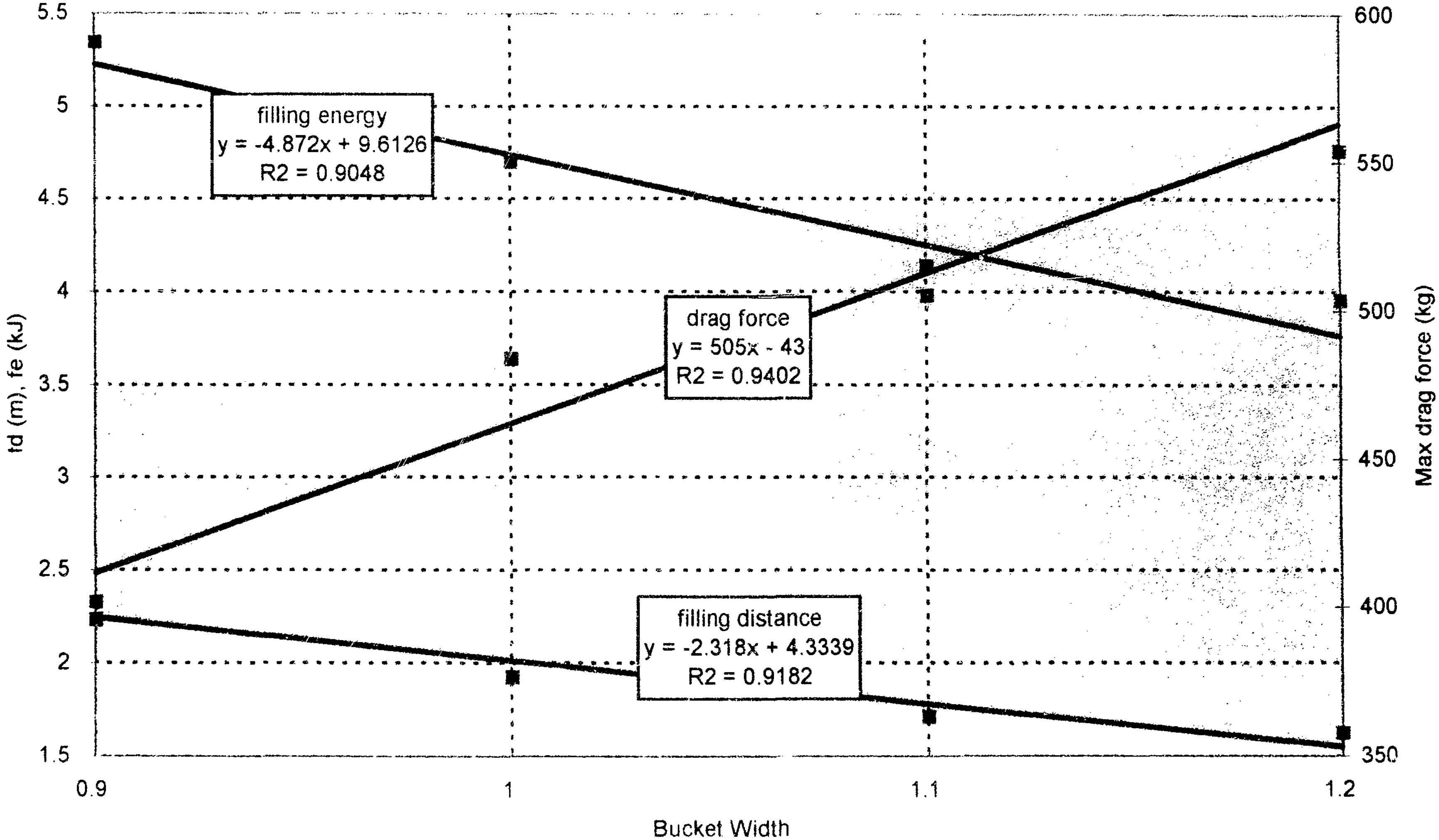

Figure C42: Relationships for performance parameters as function of bucket width on hitch height 3 in crusher run

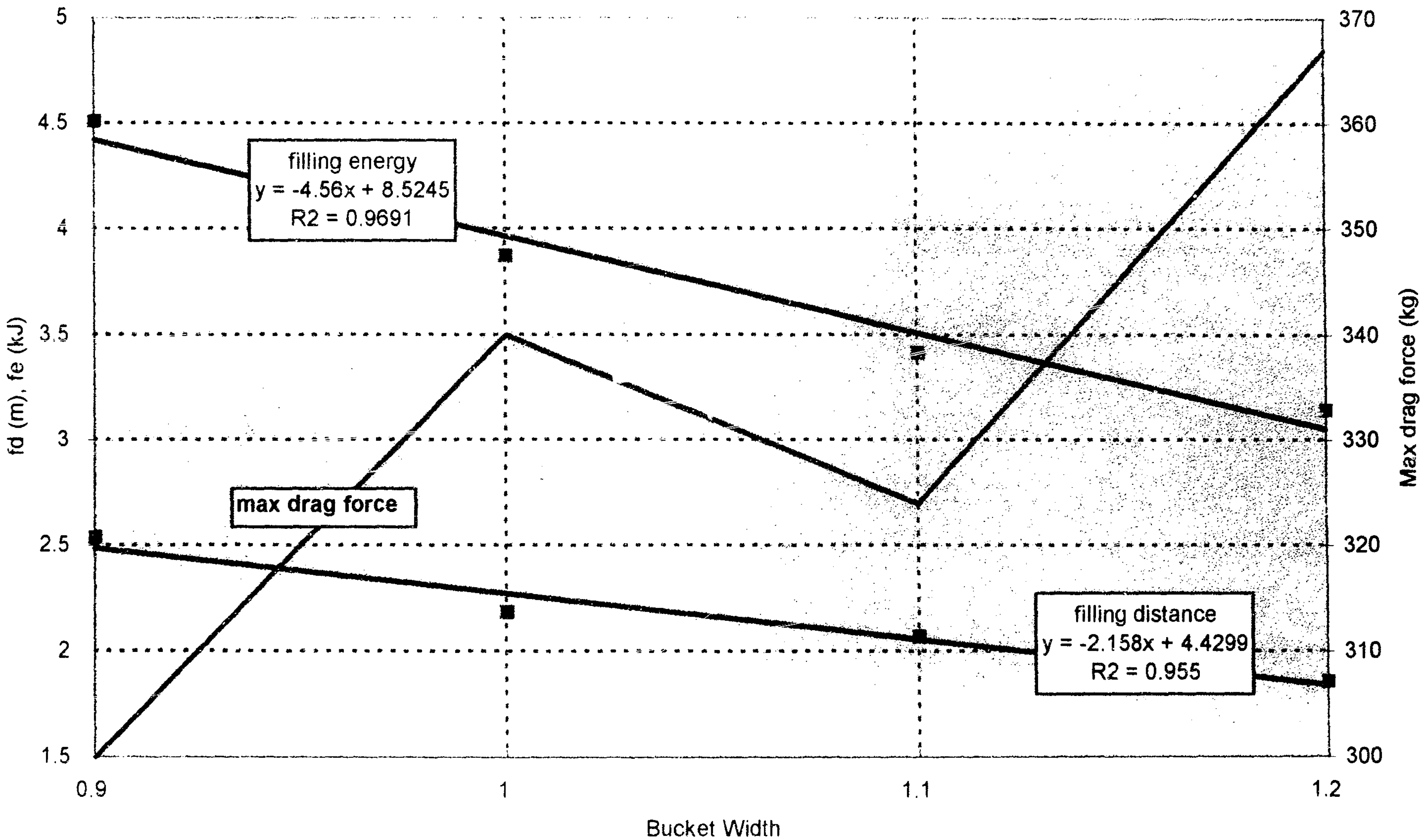

Figure C43: Relationships for performance parameters as function of bucket width on hitch height 2 in crushed rock

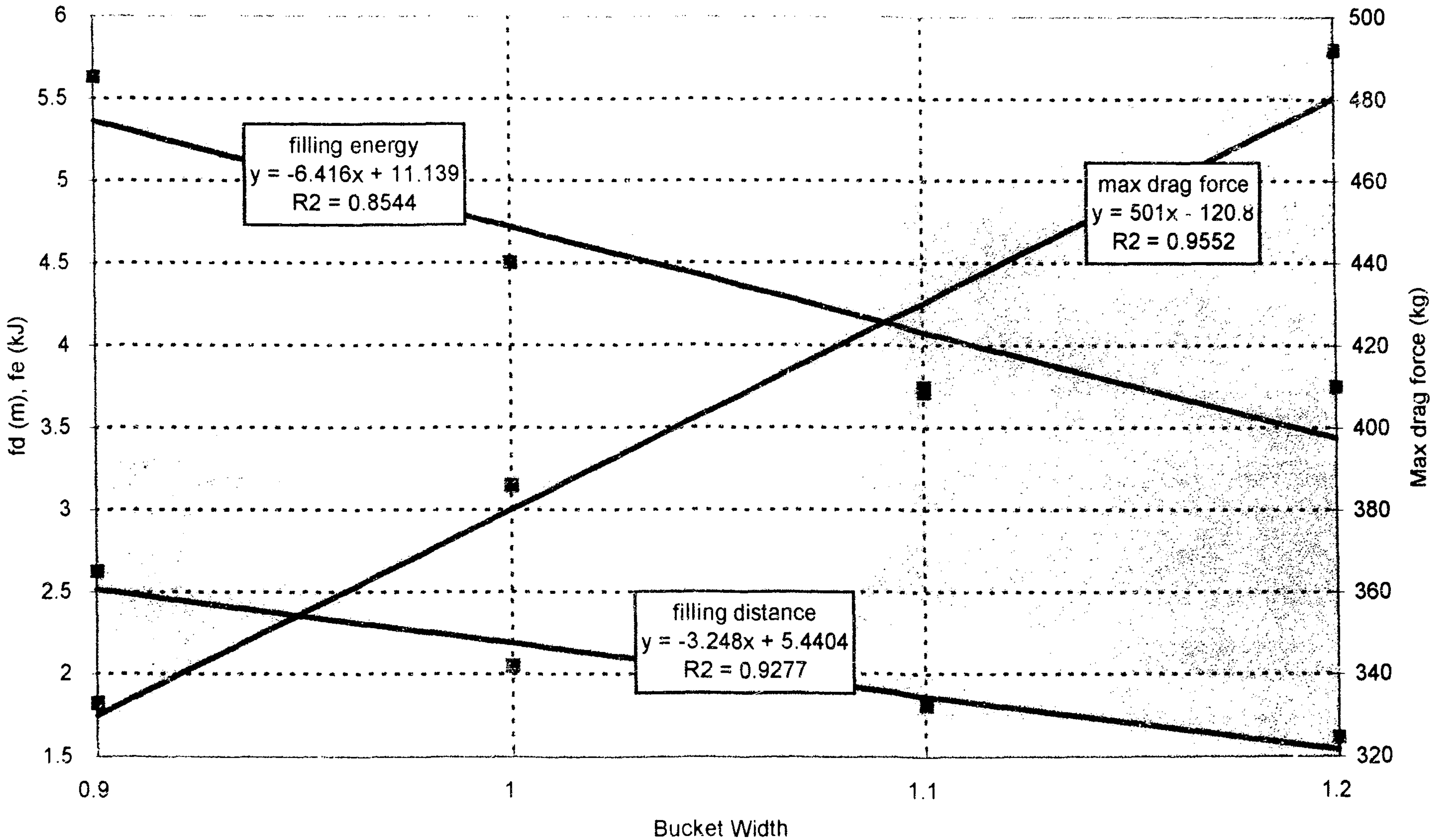

Figure C44: Relationships for performance parameters as function of bucket width on hitch height 2 in crusher run

C44

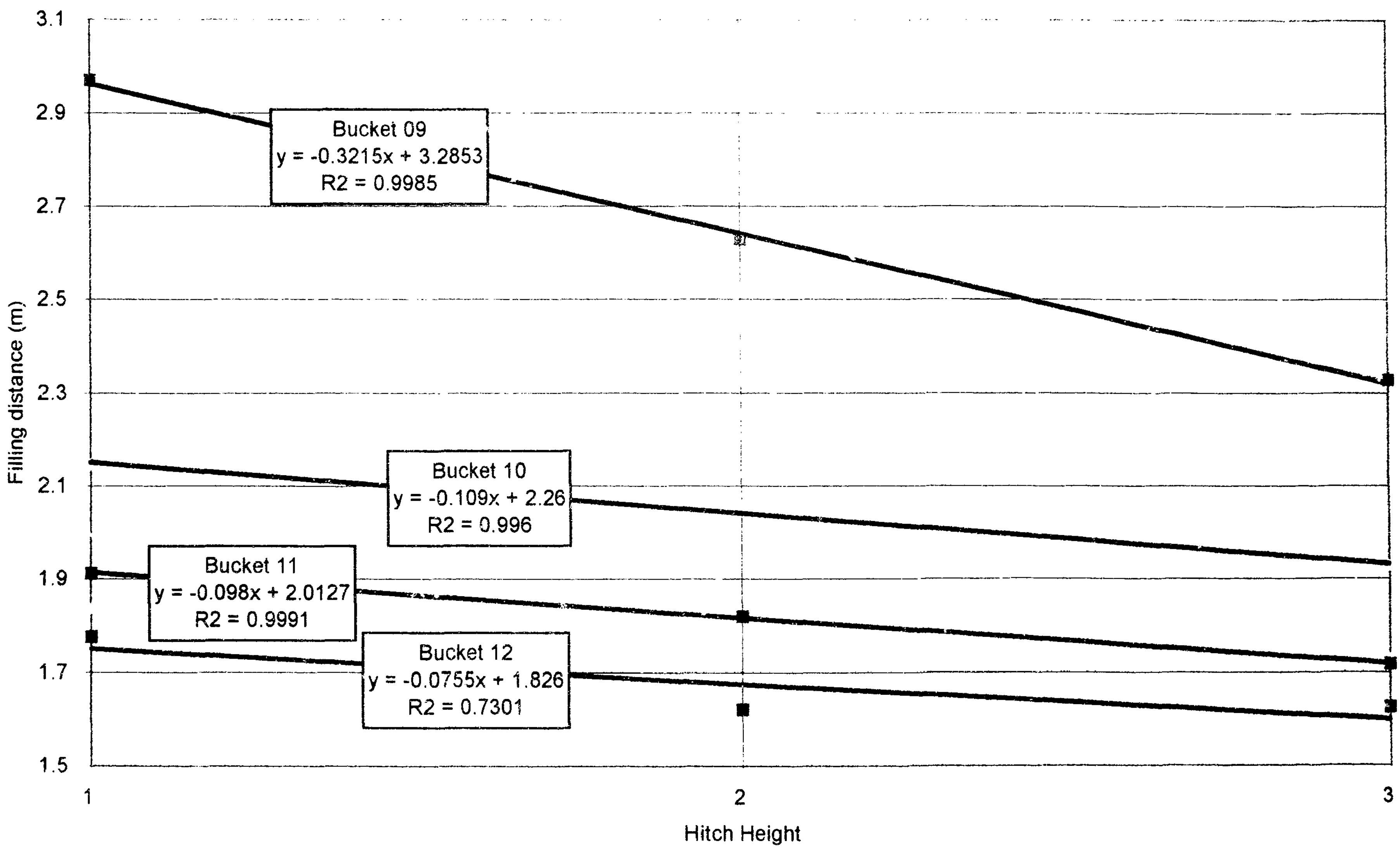

Figure C45: Relationship between Hitch Height and Filling distance for the different buckets in crusher run.

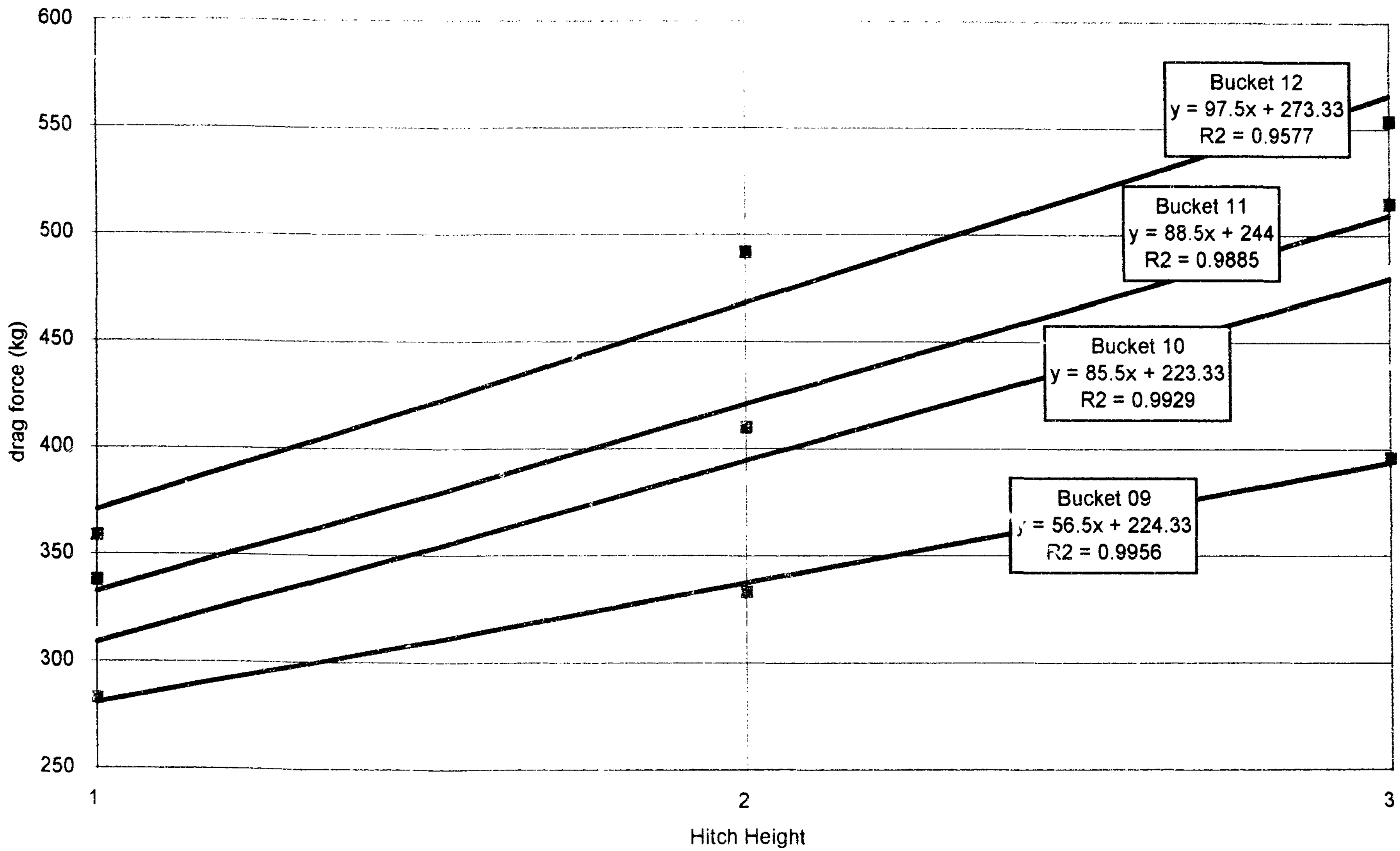

Figure C46: **Relationships for the Max drag force as a function of Hitch Height for the different buckets in crusher run.**

APPENDIX D

TEST DATA

	DRAG ANGLE=20 DEGREES			
	FD (m)	E (J)	Fmx (kg)	SDE (J/l)
CHANGE BUCKET: CRUSHED STONE				
RK12681	1.996	3121	283	55.75
RK12682	1.86	3144	367	56.92
RK12683	1.761	3095	396	57.24
RK12684	1.979	3708	416	66.56
RK12685	2.238	4003	400	72.66
RK11681	2.199	3325	276	56.46
RK11682	2.07	3416	324	58.03
RK11683	2.024	3671	413	64.69
RK11684	2.024	3911	433	68.91
RK11685	2.391	4288	408	71.35
RK10681	2.311	3791	314	63.39
RK10682	2.185	3875	340	63.39
RK10683	2.049	3934	405	66.52
RK10684	2.156	4142	423	69.24
RK10685	2.696	4991	405	82.7
RK09681	2.856	4654	277	74.8
RK09682	2.541	4511	300	72.95
RK09683	2.467	4425	339	72.41
RK09684	2.573	4815	403	76.93
RK09685	3.028	5836	401	93.29
CHANGE ANGLE OF ATTACK				
RK10661				
RK10662	2.374	3936	300	67.77
RK10663	2.131	3847	353	65.22
RK10664	2.047	3948	423	68.44
RK10665	2.421	4598	437	75.72
RK10681	2.311	3791	314	63.39
RK10682	2.185	3875	340	63.39
RK10683	2.049	3934	405	66.52
RK10684	2.156	4142	423	69.24
RK10685	2.696	4991	405	82.7
RK10601	2.41	4040	295	67.17
RK10602	2.202	3824	326	64.35
RK10603	2.083	3911	368	67.27
RK10604	2.14	4248	424	72.55
RK10605	2.718	5165	400	84.72
RK10621	2.412	3999	300	66.49
RK10622	2.039	3769	347	64.83
RK10623	2.103	4060	418	70.74
RK10624	2.274	4457	422	74.52

RK10625				
CHANGE NUMBER OF TEETH				
RK10781	2.385	3996	297	66.34
RK10782	2.099	3863	341	65.48
RK10783	2.008	3978	412	67.92
RK10784	2.194	4328	434	75.4
RK10785	2.517	5015	423	87.05
RK10721	2.203	3824	321	63.25
RK10722	2.057	3805	360	63.59
RK10723	2.063	3805	434	65.36
RK10724	2.285	4505	432	78.73
RK10725				
RK10681	2.311	3791	314	63.39
RK10682	2.185	3875	340	63.39
RK10683	2.049	3934	405	66.52
RK10684	2.156	4142	423	69.24
RK10685	2.696	4991	405	82.7
RK10621	2.412	3999	300	66.49
RK10622	2.039	3769	347	64.83
RK10623	2.103	4060	418	70.74
RK10624	2.274	4457	422	74.52
RK10625				
RK10581	2.448	3940	275	65.15
RK10582	2.144	3749	319	63.56
RK10583	2.057	3877	396	66.74
RK10584	2.139	4060	438	70.76
RK10585	2.505	4665	420	80.97
RK10521	2.434	3869	293	65.62
RK10522	2.051	3663	340	62.55
RK10523	2.041	3830	410	66.85
RK10524	2.082	4166	430	71.83
RK10525	2.648	5091	410	85.12

	DRAG ANGLE=20 DEGREES			
	FD (m)	E (J)	Fmx (kg)	SDE (J/l)
CHANGE BUCKET: CRUSHER RUN (FINES)				
CR12681	1.777	3568	359	
CR12682	1.622	3755	492	
CR12683	1.626	3959	554	
CR12684	1.733	4442	551	
CR12685	2.[illegible]85	5212	447	

CR11681	1.913	3514	338	
CR11682	1.82	3720	410	
CR11683	1.717	3984	515	
CR11684	1.782	4347	524	
CR11685	2.323	5366	450	
CR10681	2.147	4177	313	
CR10682	2.05	4502	386	
CR10683	1.929	4701	434	
CR10684	2.038	4907	505	
CR10685	2.439	5223	467	
CR09681	2.971	5796	283	
CR09682	2.628	5633	333	
CR09683	2.328	5344	396	
CR09684	2.315	5579	463	
CR09685	2.991	6848	435	
CHANGE ANGLE OF ATTACK				
CR11621	1.856	3617	374	
CR11622	1.74	3804	484	
CR11623	1.77	4113	558	
CR11624	2.107	5018	519	
CR11625				

	DRAG ANGLE = 30 DEGREES			
	FD(m)	E(J)	Fmx(kg)	SDE
RK12681	1.819	2926	310	45.73
RK12682	1.567	2790	364	43.43
RK12683	1.512	2745	393	46.43
RK12684	1.652	3178	450	51.21
RK12685	1.806	3577	427	58.59
RK11681				
RK11682				
RK11683				
RK11684				
RK11685				
RK10681	1.982	3257	279	51.14
RK10682	1.795	3154	307	49.41
RK10683	1.635	3205	380	49.07
RK10684	1.72	3404	405	52.64
RK10685	2.006	3978	393	62.14
RK09681				
RK09682				
RK09683				

RK09684				
RK09685				

	CHANGE HITCH HORISONTALLY			
	FD(m)	E(J)	Fmx(kg)	SDE
MOVE HITCH 30 MM FORWARD				
CR11681	2.089	3829	324	
CR11682	1.791	3665	386	
CR11683	1.698	4001	495	
CR11684	1.734	4209	512	
CR11685	2.094	5067	483	
HITCH IN NORMAL X-POSITION				
CR11681	1.913	3514	338	
CR11682	1.82	3720	410	
CR11683	1.717	3984	515	
CR11684	1.782	4347	524	
CR11685	2.323	5366	450	
MOVE HITCH 30 MM BACK				
CR11681	2.123	3695	336	
CR11682	1.902	3749	398	
CR11683	1.676	3714	500	
CR11684				
CR11685				

GLOSSARY

Term	Definition
angle of attack	The acute angle between the forward face of the teeth and the horizontal. It can also be defined for the lip instead of the teeth.
angle of repose	The acute angle between the side of a free poured pile of overburden and the horizontal.
bench height	Heigth above coal at which dragline is positioned.
blasting ratio	The volume (cum) of rock broken per unit weight (kg) of explosive.
block length	Length between major digout cycles.
block width	Width of each consecutive strip that is mined.
carry angle	The acute angle between the bucket floor and horizontal.
COG	The centre of gravity of the bucket.
cut length	See **block length.**
cut width	See **block width.**
chopping	A digging method in which the bucket is dropped teeth first into the overburden. This is an unproductive and wear intensive digging method.
drag angle	The acute angle between the drag ropes and the horizontal.

drag force	The total force applied to the drag ropes to overcome the resistive forces during digging. The drag stall force is the maximum that can be applied by the dragline.
dragline reach	See **dumping radius.**
dugout length	See **block length.**
dumping radius	The horizontal distance from the center of the tub to boom point.
effective reach	Horizontal distance from the edge to boom point of dragline.
filling distance	Distance that the bucket takes to fill measured along the drag rope.
filling energy	Integral of the force @ distance graph over the filling length.
hitch height	Vertical distance from bucket floor to drag hitch.
keycut	The cut that is made first in every block. It is the cut furthest from the spoil pile and is made to determine the slope and orientation of the new highwall.
panel width	See **block width**
payload	The volume (or mass) of overburden in the bucket after disengagement.
pit width	See **block width**
rehandle	Overburden that has been excavated and must be moved again.

rigging The rope and chain system that determines the orientation of the bucket when not engaged and that allows the bucket to be manipulated.

RSL Rated suspended load, the maximum weight that the dragline boom are designed to support when hanging from the boom point.

set length See **block length.**

SDE Specific digging energy, the energy required to excavate a unit volume of overburden.

stand-off Distance from the center of the dragline tub to the edge.

strip width See **block width**

struck volume The volume that the bucket would contain if filled completely from the front of the lip backwards, without any material

swell Ratio of the densities of the prime and blasted overburden.

tilt angle The acute angle between the bucket floor and the slope on which the bucket is dragged.

overburden The soil and rock that covers the coal seams.